SCREW PROPELLERS

and ESTIMATION *of* POWER *for* PROPULSION *of* SHIPS. *Also* AIR-SHIP PROPELLERS

BY

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VOL. I.—TEXT
VOL. II.—ATLAS

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AUTHOR'S PREFACE

IN 1901, while serving as an Assistant in the Bureau of Steam Engineering, United States Navy Department, I was requested by the Engineer-in-Chief, the late Rear-Admiral George W. Melville, United States Navy, to prepare a paper on the performances of the screw propellers of naval vessels.

During the preparation of this paper I became so interested in the subject that I have continued my study of it up to the present day.

From time to time, as points of interest have been developed pointing to proper lines to follow in designing screws, papers have been prepared and published in the *Journal of the American Society of Naval Engineers*.

The work here submitted is a composite of these various papers, eliminating from them all such statements and deductions as later study has demonstrated to be erroneous.

In developing the theory of design set forth in this work, the model tank trial curves of model hulls were supplied by Naval Constructor David W. Taylor, United States Navy, and the work is based upon these curves, Froude's theory of the propeller as developed by Mr. S. W. Barnaby in his work on "Screw Propellers," and the data of trials of actual vessels as supplied me by the Bureau of Steam Engineering, and I desire this work to be to them an expression of my appreciation of the aid rendered me. My thanks are also due to Mr. Luther D. Lovekin, who prepared the chapter on the geometry and draughting of propellers.

It is hoped that the book may be found to be of such value

that the words of an eminent engineer, "Any man can design a good propeller, but it takes an exceptionally fine engineer to design a bad one," will be modified in that even the exceptionally fine engineer will not be excluded.

Very sincerely,

C. W. DYSON.

PREFACE TO SECOND EDITION

WHEN the first edition of this work was presented to the engineering world, it met with very generous criticism from all sources but one, the tone being one of general approval. The one dissenting critic, writing for the London "Shipbuilding and Shipping Record," made some caustic statements concerning the work, and the opportunity is now grasped to inform him that later experience has demonstrated that every criticism made by him was a just one.

In the first edition were two glaring faults, one being the vagueness of the method for determining the thrust deduction factor and the other, the method of applying it in determining the characteristics of the propeller. These have both been eliminated to a great degree by a more thorough study of the effects of hull form and position of the propeller in relation to the hull on the performance of the propeller. The approximate method of computation called "the method of reduced diameter" has been replaced by the more accurate method of "variation in load."

The Charts of propulsive coefficients and Tip Speeds have been replaced by the equations derived for the "law of varying load" and the "law of varying power and speed."

The author's ideas concerning the phenomenon of cavitation having become crystallized during the later years through more thorough investigation, they are now presented in the chapter devoted to that subject.

A chapter dealing with the design of the aeroplane propeller has also been added, but this can not be regarded as of nearly the same accuracy as that part of the work devoted to hydraulic propellers as actual measurements of powers, revolutions and thrusts occurring in actual flight are missing, and until such

data are available, design curves and factors of absolute accuracy can not be obtained.

The author having carried his work on propellers as far as he feels able, will now lay it down, trusting that it will be picked up by younger and more energetic hands, who, loving the subject to the same extent, will carry the work along until "the last word on propellers" has been said.

In making his final bow, the author desires to express his thanks to:

Engineer-in-Chief Robert S. Griffin, U. S. Navy, and Chief Constructor David W. Taylor, U. S. Navy, for the practical aid and encouragement they have given him throughout the past many years. The large shipbuilding companies of the United States for their generosity in providing him with data of performances of vessels. The Marine Architects and Engineers of the United States for the praise and encouragement in many forms that he has received from them.

Mr. Spencer Heath of the American Propeller and Manufacturing Company of Baltimore, Md., to whom the author is indebted for that part of the work devoted to materials for and details of construction of aeroplane propellers.

Lieutenant Commander S. M. Robinson, U. S. Navy, who has been of the greatest assistance in the prosecution of the work.

The engineering press that has been extremely generous in devoting its columns to encouraging notices of the author's endeavors. The propeller expert of the London "Shipbuilding and Shipping Record," whose criticisms concerning the first edition of the book spurred the author on to renewed investigations.

The publishers for their kindness in offering an opportunity to present the subject matter in an enduring form.

And to the kind fate which led the author into a line of work from which he has derived an enormous amount of pleasure for seventeen years, and which located him in a position where this line of work could be successfully carried out.

Very sincerely,

C. W. Dyson.

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SCREW PROPELLERS

INTRODUCTION

A SHORT HISTORY OF THE DEVELOPMENT OF SCREW-PROPELLER PROPULSION

John Bourne, in his "Treatise on the Screw Propeller," published in 1867, states that "the screw propeller is, in all probability, a very ancient contrivance. In China it is said to have been known for ages; but in European countries the idea of a screw propeller appears to have been derived either from the windmill or smoke-jack, or from the screw of Archimedes, an instrument much used in some countries for raising water."

Seaton, in his work on screw propellers, traces its development from the time when man first used his hands as paddles, through the pulling oar and the sculling oar, and the modified application of the latter in the form of the screw propeller.

These suppositions and tracings of lineage are very interesting to read and consider, but it hardly appears necessary to delve so deeply in order to understand why this form of propulsion exists and how it originated. To any people who were acquainted with the principle of the screw thread working in a nut, and who were looking for a means of decreasing the labor necessary in propelling their marine craft, the screw propeller would appear to be the rational application of the screw thread for this purpose, as the oar, and, later, its rational successor, the paddle-wheel, were of the lever and fulcrum.

The idea of making a screw on the plan of a windmill to work in water appears to have originated in England with Robert Hooke, one of the most remarkable men that country has ever produced.

He proposed the idea in 1681, in a work entitled "Philosophical Collections," but it remained an idea only until 1731, when a Monsieur Du Quet invented a contrivance for dragging vessels up against a stream by means of a screw, or helical feather, which is turned around by the water.

Du Quet was followed in 1746 by Bougner, who proposed to employ revolving arms, like the vanes of a windmill; but this scheme, it is stated, had not been found to possess sufficient force.

In 1768 Pancton, in 1776 Bushnell, an American, and in 1785 Bramah, all proposed various means of applying the screw propeller, the latter's proposal being notable from the fact that he was the first who proposed to fix the screw at the stern "in or about the place where the rudder is usually placed," to be worked by a shaft proceeding direct from the engine.

The first application of the screw propeller to an actual vessel, of which we have any record, was made by William Lyttleton in 1794.

The propeller consisted of three helical feathers wound on a cylinder, and these cylinders were to be so fixed at the bow and stern, or at the sides, as to be immersed in the water, and to carry the vessel forward when put into revolution. Each cylinder, or screw, was to be turned by an endless rope working in a sheave.

Upon trial, the effect of the screw was much less than expected, a speed of only two miles an hour being obtained. This invention was said to have been brought from China.

In the years that followed, up until 1816, several inventions were made and experiments tried, but with little success. In 1816, Robert Buchanan, in a work on steam propulsion of vessels, in writing of the screw propeller, stated that "some mechanics, however, still think favorably of it, and suppose that if a screw of only one revolution were used, it would be better than where a longer thread is employed." Experience has since amply demonstrated that this proposed restriction of the length of the screw was founded upon just views.

Another period of years passed during which several inven-

tions of screw propellers were made, but of which no trials occurred. In 1824 and 1825, Dollman, a Frenchman, and Perkins, an Englishman, both proposed " two concentric axes turning in opposite directions, and each bearing two blades, are placed at the stern of the vessel, and by the revolution of the blades in opposite directions the vessel is propelled."

We see this type of propeller in use to-day for torpedo propulsion.

In this latter year, 1825, a company which had been formed for carrying into operation a project of a gas vacuum engine offered a reward for the best suggestion for propelling vessels without paddle-wheels. The reward was gained by Samuel Brown, the inventor of the engine, who proposed to accomplish the desired object by a screw placed in the bow of the vessel. A vessel was built and fitted with a screw; and with this vessel a speed of six or seven miles an hour is said to have been attained.

As the primary object of the experiment was to introduce the gas vacuum engine, and this engine having failed, the propeller was given practical credit for the failure, the company was broken up, and the scheme abandoned.

In 1827, Tredgold indicated the desirability of making screws with an expanding or increasing pitch. He stated, " if it (the spiral) be continued, it should be made with a decreasing angle," because during the first revolution of the spiral the water would have obtained all the velocity the spiral of the original angle could communicate.

In 1830, Josiah Sopley, an American, proposed a propeller of eight or any other number of vanes, these vanes to form " segments of spirals."

In 1836, John Ericsson patented an improved propeller applicable to steam navigation. This propeller consisted of two thin broad hoops, or short cylinders, made to revolve in contrary directions around a common center, each cylinder or hoop moving with different velocity from the other; such hoops or cylinders being also situated entirely under the water at the stern of a boat, and furnished each with a series of short spiral planes of plates—the plates of each series standing at an angle the exact

converse of the angle given to those of the other series, and kept revolving by the power of a steam engine.

In some cases, Ericsson made use of two screws, one behind the other; in others, of one screw on each quarter, but generally he used a single screw of a number of threads placed before the rudder in the stern. This propeller is stated to have been very successful and efficient.

In 1839, a Mr. Baddeley stated that many years before that time a Mr. Weddell had fitted a vessel with a propeller with which he had made a voyage to Africa. This is the first record we have of the propeller being used for deep-sea work and long voyages. The conclusion arrived at after this trial was that paddle-wheels of large diameter and little dip had greater propelling efficiency than a screw.

The first successful operation of the screw as a propeller, however, may be considered to have occurred with that of Ericsson in 1836, and with that of Smith in 1839, the latter having fitted a screw consisting of a single-threaded helix of one complete convolution to a vessel of 237 tons burden named the *Archimedes*. A double thread of half a convolution was afterwards tried, and found to be an improvement, but the best result was obtained with two threads and one-sixth of a convolution.

The first use made of the screw propeller by the British Navy was in 1802, when a propeller invented by a man named Shorter was tried on board H. M. S.'s *Dragon* and *Superb* and on the transport *Doncaster*. This latter ship attained a speed of 1½ miles per hour when deeply laden, with eight men only at the capstan which worked the screw.

No further use of the screw propeller was made in the British Navy until 1843, when H. M. S. *Rattler* was completed. This vessel developed a speed of ten knots.

In 1845, the first screw steamer, the *Great Britain*, crossed the Atlantic.

The first vessel in our navy to be fitted with a screw propeller was the *Waterwitch*, in 1845 or 1846, followed by the *Allegheny*, in 1852. Both these vessels were originally fitted with paddle-wheels (horizontal submerged), the invention of Lieu-

tenant Hunter, of the navy, and had proven failures. Their engines being better adapted to driving screw propellers than for paddle-wheels, screw propellers were adopted in place of the Hunter wheels.

Since the general adoption of the screw propeller for marine propulsion, it has proven an exceedingly attractive field for the inventor. The number of inventions and patents that have been taken out covering every individual item of the instrument is myriad, and one can hardly suggest anything concerning any part of the propeller which he would not find had already been suggested or patented by some one in the past.

For many years after its adoption for the propulsion of ships the seeming vagaries in the performances of screw propellers in actual service of propulsion cast a great mystery over it and over the laws governing its action.

The greater part of this mystery is, however, not due to the propeller, but can be directly attributed to the carelessness with which trials of ships have been conducted and with which the data of performances have been collected. The major part of the remainder of the mystery is due entirely to the effect of variations in hull form with the changing character of the flow of water to the propeller accompanying these variations, and the resultant effect on the propulsive efficiency; and to incorrect estimates of effective horse-powers required for given speeds, these estimates of power having been based on frictional and residual resistances of the bare hull of the vessel, the malign influence of the appendages fitted to the hull not having been appreciated and, therefore, having been entirely neglected.

The small residue of the mystery can be ascribed to the propeller itself, and is partly due to the myriad variations in blade forms and sections which have been used, these apparently depending upon the taste of the individual designer; and finally, to the lack of a consistent basis of comparison by which the performances of screw propellers could either be analyzed or predicted with any degree of certainty.

As the years rolled by they brought in their wake the model tank by means of which a more nearly accurate value of the

effective horse-power required for any desired speed of any given hull can be obtained; more accurate instruments for the measurement of indicated and shaft horse-powers; better mechanical construction of propelling engines by which mechanical efficiency has been greatly increased and brought to a more nearly constant value; machining of propellers to designed diameter, pitch and area, thus fixing more definitely the most important characteristics, and reducing the frictional losses of the propeller; more care in conducting trials over measured courses combined with a better knowledge of the effects of shallow water and varying currents on such courses.

All of these improvements have resulted in the production of data of such accuracy that curves may be laid down, based on these data, by means of which the performance of any given propeller can be analyzed or predicted or by which a propeller correctly proportioned for any given conditions can be designed.

CHAPTER I

BLOCK COEFFICIENT, THRUST DEDUCTION

In erecting the column representing screw propeller design, the stones that form the foundation are the stone of hull form and the stone of effective horse-power to be delivered. The other stones necessary to complete the column are those of diameter, pitch, revolutions and projected area ratio. These form the principal stones entering into the structure; in addition there are also required the minor, less important ones called blade form and blade section.

All of the above stones are so formed as to interlock and any variation in one of them necessitates a change in every one of the others in order to preserve the form and stability of the completed column.

The column when completed may be called the "Column of Propulsive Efficiency," and in studying the different stones entering into it, those forming the foundation will be considered first.

Derivation of Block Coefficient to Use in Calculation of Propeller

Should there be adopted for different classes of vessels standard sets of bow and stern lines and standard shapes of midship sections, there would be for all vessels of any class, no matter what the ratio of Beam to Length on the Load Water Line nor what might be the length of the middle body of the ship, a constant condition of circumstances governing the flow of water to the propellers.

It would also be found that the nominal block coefficients$\{=35\times$Displacement$\div$(Beam$\times$Length on Load Water Line$\times$Draught)$\}$, would change, approximately, inversely with

the ratio Beam ÷ Length on Load Water Line, while the actual hull conditions, so far as affecting the propeller performance, would remain constant.

Bearing these facts in mind, and having adopted a standard series of block coefficients and Beam ratios, it becomes necessary to lay down a guide chart for the determination of the standard block coefficient corresponding to any block coefficient, beam ratio and midship section coefficient, also to correct the resultant block for variations in positions of propellers from the standard positions of the basic vessels.

Sheet 17.—For Correction of Block Coefficient

On this chart the abscissas are Block Coefficients, while for the Block Corrections the ordinates are values of Beam ÷ Length on L.W.L. For checking the Block by means of the Coefficient of Immersed Midship Section, the ordinates are Coefficients of Immersed Midship Section. All vessels, whose Immersed Midship Section coincides with the Standard Curve, will be of standard fore- and after-body (abnormal designs of hull not being considered). Those plotting below the Standard Curve will be bluffer, forward and aft, and those plotting above the curve will be finer than the Standard Hulls.

Use of Sheet 17 in Propeller Design

In the lower section of this Chart are shown three diagonal lines, X, Y, and Z. Line X is for the Standard vessels from which the Charts of propeller design were obtained. Such vessels have coefficients of Immersed Midship Section falling close to the curve of M.S. coefficient marked Standard, have propellers located well clear of the hull so that loss, through interference of flow of water to the screw by the hull, is a minimum.

As the location of the screws draws closer in behind the hull, and the influence of the wake has sensibly increased over that of condition X, the line Y takes the place of X. Where the propeller is located close to and directly to the rear of the stern post,

so that the full wake effect of the hull is encountered, line Z replaces both X and Y in the determination of the Block Coefficient.

To apply this Chart, let us suppose we have three vessels, A, B, and E, having the following characteristics:

	A	B	E
Nominal Block Coefficient.............	.78	.665	.61
Beam ÷ Length on L.W.L.............	.123	.186	.217

It is required to find the Block Coefficient to use in the design of the propeller and also to estimate for the expected appendage resistances.

Plot A, B, and E, on the chart with the nominal block coefficients as abscissas and with the values of Beam ÷ L.L.W.L. as ordinates. Through these plotted points and the unity abscissa point, pass a straight line, extending it until it cuts line X. In the cases taken, A, B, and E are all on the same line passing through the unity value of abscissas. Where this line crosses X, at B, project up to the Standard curve of midship section coefficient. Should the M.S. coefficient of the vessel in question plot near to the Standard curve of M.S. coefficient, the vessel's ends may be considered standard, and the vessel's block coefficient be taken as that given by the abscissa value of B. Should it fall above this curve, that is, the M.S. be fuller than standard, while the vessel plots at a value of B ÷ L.L.W.L. below X, the ends will be finer; if below, fuller than standard, and the block coefficients be modified accordingly, that is by multiplying the standard B.C. by the inverse ratio of the midship section coefficients, unless the vessel be one having a nominal block coefficient of not less than .5 and the after body be very fine, in which case the correction for variation of midship section should not be made. Should the propellers be located in the conditions given by lines Y or Z the fineness will be gauged, as before, by the intersection of the line with X, but the actual block to use for the propeller design will be that given by the abscissa value of the point of intersection with Y or Z, except where correction is made for variation from the Standard M.S. Coefficient.

The intersections of the cross line through D with the standard lines X, Y, Z, may be found mathematically as follows:

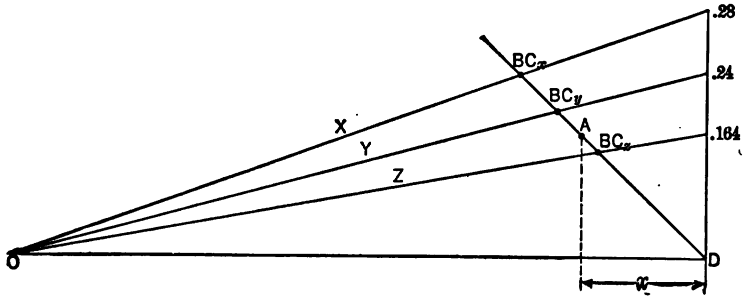


FIG. 1.—Diagram foı Computing Slip Block Coefficient.

Vessel A – nominal block coefficient $= 1 - x =$ B.C.$_n$.

$$B \div \text{L.L.W.L.} = y.$$

$$\text{B.C.}_x = \frac{y}{.28(1 - \text{B.C.}_n) + y};$$

$$\text{B.C.}_y = \frac{y}{.248(1 - \text{B.C.}_n) + y};$$

$$\text{B.C.}_z = \frac{y}{.164(1 - \text{B.C.}_n) + y}.$$

The above gives approximate block coefficients to use with charts of design, but makes no allowance for variation of form of immersed midship section from standard form.

It may be used when Sheet 17 is not available for graphic correction of block coefficient.

While, in general, the slip block coefficients should be obtained as above described, there are cases, however, where the method of estimating the slip block coefficient should depart from this method. These cases are three in number, the first of which has already been given but is here repeated:

1. The vessel has a nominal block coefficient of not less than .5, and a midship section coefficient much finer than standard as given by Sheet 17; the propellers located in condition 3,

Sheet 19. No correction should be made for variation of midship section from standard.

2. Single screw ships of beam very broad as compared with length and draft of vessel. In such vessels the immersed lines of the vessel correspond to the lines below the turn of the bilge in vessels of orthodox form. No correction should be made for midship section variation. Such vessels are similar to shallow draft ferry boats. In estimating the slip block coefficient of single screw vessels, the condition of immersed hull body as to nominal Block Coefficient existing when the tips of the upper vertical blades are immersed to a depth of about 15 per cent of the diameter of the screw should be used.

3. Single or twin screw tunnel boats. These vessels have the propellers so located that the only definite idea of the flow of water to the propeller that can be obtained is that of its direction and this may be considered as normal to the disc. The only thrust deduction loss that occurs is that due to friction in the tunnel and amounts to $K=1.195$ and this can be considered as constant. Make no corrections in obtaining the slip block coefficient, but use the nominal block coefficient for the slip block coefficient.

The lines X, Y and Z may be called "orthodox" for usual types of hulls and location of propellers. There are, however, many departures from these "orthodox" conditions and each of these departures produces a change in wake and, therefore, a change in revolutions of propeller for given powers of engines and speeds of vessel.

These departures may be classed under four separate heads, as follows:

1. Deep draft vessels fitted with propellers of diameters bearing a ratio of less than .70 to the draft, the lower blades passing close to or below the keel.

In such a case, with the vessel running at light draft with the propeller diameter bearing a large ratio to the light draft, but entirely submerged, the slip block coefficient will be the normal from line Z corresponding to the L.L.W.L. or L.B.P., the beam B, and the displacement at the light draft. In estimating the

apparent slip, the value Log A, must be taken from the curve marked X on Sheet 21. See Par. 2.

As the vessel is loaded and the draft increases, conditions of wake change very slightly, and may be neglected, until after passing a ratio of diameter to draft, D : H of .75. Shortly after passing this ratio the wake rapidly reduces until D : H = about .70, when the wake is that corresponding to the S.B.C. as taken from line W, Sheet 17, while the value of Log A, must be taken from the curve X, Sheet 21. The thrust-deduction factor (K, see next section), will be that corresponding to the S.B.C. of the light draft condition.

2. Shallow draft vessels, 14 ft. and less, of S.B.C. = approximately .8 and greater. The S.B.C. will be that corresponding to Line Z, but the value Log A, must be taken from the curve Y, Sheet 21. This applies to single screw ships.

3. Submarines of the Lake type, by which term is meant all submarines carrying their propellers beneath the hull, either single screw or twin. Such vessels when working on the surface should have their S.B.C. taken off from the line T, when trimmed by the stern and from V for even keel, while when submerged, the S.B.C. should be taken from the line V. The nominal block coefficient to use with $B \div$L.L.W.L, being that of the surface condition. In both surface and submerged conditions, however, the value of Log A, should be taken from Curve Y, Sheet 21.

4. Submarines of the Holland type, by which term is meant all submarines carrying their propellers abaft and clear of the hull. For both surface and submerged conditions the S.B.C.'s should be taken from line U, the nominal B.C. being that corresponding to the surface condition. The value log A, for the surface condition should be taken from Curve X, Sheet 21, and for the submerged condition from Curve Y of this same sheet.

5. Very fine vessels of high speed where heavy squatting occurs, have the slip B.C. taken from Line X, Sheet 17, but after reaching a certain amount of squat, the value log A, gradually passes from Curve X, Sheet 21, to Curve Y. In the case of destroyers where the propellers are located abreast the stern

post, or where the propellers are several feet forward of the stern post and the axes of fore-and-aft sections of the lower strut arms are inclined downward from aft forward to bring them into the stream lines, the departure from Curve X, Sheet 21, begins at about $v \div \sqrt{\text{L.L.W.L.}} = 1.48$ and reaches Y when

$$v \div \sqrt{\text{L.L.W.L.}} = 2.13.$$

These five conditions are extremely important in their bearing on revolutions and should be thoroughly borne in mind.

Thrust Deduction and Wake Gain—Sheet 18

When a propeller works at the stern of a vessel it operates in a body of water which partakes, in a more or less degree, of the forward motion of the vessel. When the propeller is so located that the column of water entering the propeller enters normal to the propeller disc and with very little disturbance, and when, in addition, the propeller blade tips are well immersed and pass the hull at a good distance from it, the wake, as the forward motion of the water is called, will increase the effective thrust of the propeller for any given indicated or shaft horse-power which may be delivered by the propelling engine. This gain is known as the *wake gain.*

Should the propeller be so located in relation to the hull that the water entering the propeller, in place of entering normal to the disc enters at a more or less obtuse angle to that plane, or should the propeller blades be insufficiently immersed so that the propeller draws down considerable quantities of air into its suction column, or should the propeller blades with certain forms of ship's lines pass unduly close to the hull, or should combinations of these conditions exist, the effective thrust per revolution for any given indicated or shaft horse-power delivered by the propelling engine will be reduced. This loss in propulsive efficiency is called the *thrust deduction.*

The action of the water leaving the propeller is illustrated in Fig. 1*A*. Should a piece of floss thread be taken and secured

to the guard wires on the discharge side of a ventilating fan, it would be seen that the particles of air instead of leaving the fan normal to its disc, pass away in lines forming the generatrices of concentric right hyperboloids of revolution, the maximum belt diameter A being determined by the angle at which the tip currents leave the fan, and this angle being, in turn, determined by the entering angle of the currents entering the propeller or fan at the tips.

The same state of flow undoubtedly exists in the case of water flowing to a propeller, and the more nearly normal to the disc of the propeller is the direction of entry flow, the greater will

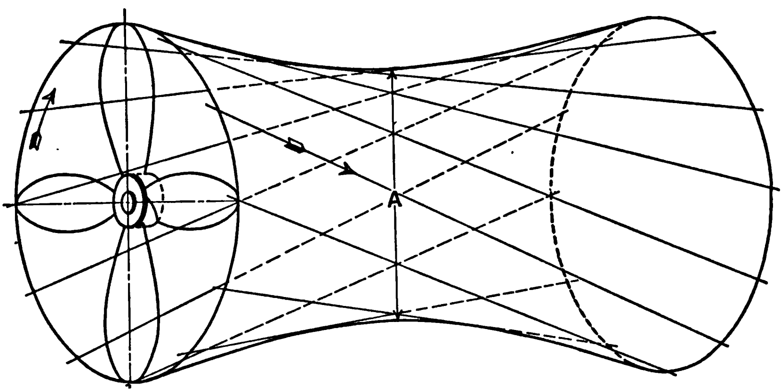


FIG. 1A.—Lines of Flow from Propeller.

be the belt area at A and the lower will be the thrust exerted at the belt per unit of area, while at the same time its direction will be more nearly in the direction of advance of the screw and the greater will be the efficiency of propulsion.

Furthermore, the more the angularity of flow and turbulence of flow occurring as the water enters the propeller, the greater will be the change of direction of flow which must occur as the water passes through the fore-and-aft length of the propeller blades. Should this length be short, the water may leave the blades before the change in direction has been accomplished and a loss in efficiency in addition to the normal thrust deduction will occur. This phenomenon occurs as the speed of advance of the

propeller for a given number of revolutions is reduced beyond a certain amount, the phenomenon being referred to later in this work as "dispersal of the thrust column." It is a comrade of the phenomenon generally known as "Cavitation," and, like its comrade, its arrival can be retarded by increasing the projected area ratio of the propeller, which carries with it an increase in the fore-and-aft length of the propeller and a greater time allowance for the propeller to swing the currents of water into the lines of efficient thrust; this retardation is, however, not obtained without a price, which, is a reduction in the propulsive efficiency at lower speeds where neither "cavitation" nor "dispersal of the thrust column" need be feared.

In cases where the thrust deduction exceeds the wake gain, and such cases are the usual ones where the standard block coefficient (slip block coefficient) for the propeller position is .55 or greater, the result is a net loss in propulsive efficiency requiring an increase in revolutions with an accompanying increase in engine power. Should the wake gain exceed the thrust deduction, the opposite effect will be produced.

Calling the percentage increase of power required by the thrust deduction loss, t, and the reduction in power caused by the wake gain, w, the resulting factor to apply to the calculated power to produce any given thrust can be represented by

$$K=(1+t-w) \text{ and where } t=w,\ K=1.$$

Control of the Value of K

As the value of K is fixed by the character of the hull lines and in certain cases by the position of the propeller relative to these lines, there may exist a slight amount of freedom in fixing the value of K for any given problem. By practical conditions which are forced upon the designer, the propeller cannot be removed farther aft from the fullness of the hull lines than a certain distance, this distance being controlled by the necessity for the shaft and propeller supports, and this maximum distance fixes the minimum value of K for any hull.

For some types of hull lines this value of K will increase as the tip clearance between hull and propeller decreases below a certain amount, this amount depending upon the slip block coefficient of the vessel and upon the height of the horizontal line of least tip clearance, usually the height of the hub center above the base line of the vessel.

The amount of immersion of the upper tips of the blades below the water surface appears to have some influence upon the thrust deduction, but very slight as compared with hull tip clearance, for vessels acting on the surface.

On Sheet 19 are shown four screw propellers illustrating positions and resultant effect on thrust deduction for different types of hulls. This same sheet also gives the variations of the thrust deduction factor K for varying slip block coefficients and tip clearances.

Where propellers are located as shown by position 1, the vertical through the hub center piercing the skin of the ship well below the surface of the water, the thrust deduction factor K increases as the relative tip clearance decreases and reaches the limit given by the lower bounding curve C_1–C_2 where it has its maximum value.

For vessels having the propellers located as shown by position 2, the vertical through the center of the propeller hub piercing the skin of the vessel well above the water line, the midship section of the vessel being standard or fuller than standard, K appears to have the values given by the curve C_1–C_2.

In cases where the propellers are located as shown by position 3, the vertical through the hub center passing entirely clear of the vessel or piercing the hull well above the water, due to fineness of midship section and of after body, the value of K appears to be practically constant for all values of relative tip clearance, the values for the different values of slip block coefficient being given by the curve C_3–C_2.

To obtain the relative tip clearance, a propeller having the center of the hub 10 ft. above the base line of the vessel, and the tip of an upper vertical blade 14 ft. above the base line are taken as reference conditions, then calling:

H^1 (Wing Screws) = (Actual horizontal tip clearance $\times 10$) $\div$ actual height of center of hub above base line.

H (Depth of Immersion of upper blade tip) = (Actual immersion in feet $\times 14$) $\div$ actual height of tip above base line.

Then

$$\text{Relative Tip Clearance} = \frac{H + 10H'}{11}.$$

Where vessels are fitted with single screws, located directly abaft the stern post, the amount of thrust deduction for any hull appears to depend upon the slip block coefficient of the hull and upon the actual mean fore-and-aft clearance between the propeller blades and the skin of the ship, thus for ships of similar blocks, the thrust deduction appears to be a function of the draft of the vessel as the fore-and-aft blade clearances will vary with the draft. The values of the thrust deduction factors for such vessels apparently reach a minimum at about 20 ft. draft, and are shown by the curve C_3–C_2, while the maximum values are reached at about 12 ft. draft and are given by the curve C–C_2. Where with these shallow draft vessels, the propeller end of the shaft line is gradually lowered as the block fulls, until at a .9 slip block nearly the full length of the lower blade extends below the keel, the thrust deduction factors follow the line C–C–C_2.

These values of K hold, however, for effective thrusts equal to or less than those corresponding to the line $E.T.$ on Sheet 22. When these critical thrusts are exceeded the value of K rapidly increases. This increase in K is, however, treated as a loss in propulsive efficiency and the percentages of the efficiency realized with thrusts $E.T.$, which can be realized with increased thrusts are shown as curves on Sheet 22. The augmentation of K produced by excess of effective thrusts over the values of the critical thrusts, $E.T.$, is expressed by $K \times \left(\frac{\text{e.t.}}{E.T.}\right)^x$, where e.t. equals the actual effective thrust for any load condition and $E.T.$ equals the critical effective thrust for the same condition.

CHAPTER II

ESTIMATION OF POWER. INDICATED, SHAFT, THRUST, EFFECTIVE (TOW ROPE) HORSE-POWER

WHEN speaking of the power required to drive any given vessel at a certain speed, it is usually referred to as the Indicated Horse-power where reciprocating engines are used for power development, and as Shaft Horse-power where turbines or some other form of rotary engine is used.

By Indicated Horse-power is meant the power developed in the steam cylinders of the engine by the steam pressure on the pistons acting through the distance travelled by them. It is calculated by means of the following equation:

$$\text{I.H.P.} = \frac{P \times L \times A \times N}{33,000},$$

where P = the mean effective pressure on the piston per stroke, in pounds;

L = Length of piston stroke in feet;

N = Number of strokes per minute;

A = Area of the piston in square inches.

There is a percentage of the Indicated Horse-power which is lost in the engine itself and in the shaft bearings due to friction of the moving parts. In this book this is taken as equal to 8 per cent. The remainder of the engine power is available for turning the propeller and is known as Shaft Horse-power—where Shaft Horse-power = S.H.P. = .92 I.H.P.

This latter power being transmitted to the propeller, the latter delivers a thrust in pushing the ship ahead, and the resulting power, called Thrust Horse-power, is measured by multiplying the actual thrust in pounds by the number of feet moved through

by the ship per minute and dividing the product by 33,000. The equation is:

$$\text{Thrust Horse-power} = \text{T.H.P.} = \frac{T \times v \times 6080}{33{,}000 \times 60} = (T \times v) \div 326,$$

and the Efficiency of the Propeller $= E =$ T.H.P. ÷ S.H.P.

The power which would be actually necessary to tow a vessel through the water at any given speed is usually referred to as the Effective or Tow Rope Horse-power, and, calling the tension on the tow-rope T_r, the equation for Effective Horse-power is:

$$\text{Effective Horse-power} = \text{E.H.P.} = \frac{T_r \times v \times 6080}{33{,}000 \times 60} = (T_r \times v) \div 326$$

and the

Propulsive Efficiency = E.H.P. ÷ I.H.P.

In the actual making of the estimate of I.H.P. or S.H.P. necessary for the propulsion of any given vessel at any desired speed, it is necessary, first of all, to obtain the proper value of the E.H.P. required for this speed.

The methods of doing this are four in number, as follows:

1. The Admiralty Coefficient.
2. The Law of Comparison.
3. Independent Estimate.
4. Model Experiments.

Of these methods, 4 is to be preferred.

Admiralty Coefficient

The equation in which this coefficient occurs is

$$\text{I.H.P.} = \frac{D^{2/3} v^3}{K_a},$$

in which

I.H.P. = Indicated horse-power of the engine;
D = Displacement, in tons;
v = Speed, in knots per hour;
K_a = Admiralty Coefficient.

Where the Shaft Horse-power is given instead of the Indicated Horse-power, the equation should read:

$$\text{S.H.P.} = \frac{D^{2/3}v^3 \times .92}{K_a}.$$

This coefficient, K_a, must be derived from some ship for which the displacement, power, and speed are known, and further, in order that a close agreement may be expected between the estimated speed and the actual trial speed of the ship, the coefficient must be derived from a ship that is geometrically similar to the ship under design, and which has the corresponding speed. These terms will be explained when the "Law of Comparison" is taken up. Furthermore, it is absolutely necessary that the conditions existing in the new ship are such as will permit the realization of an equal coefficient of propulsion with the compared vessel. Where these conditions exist, we may write

$$\text{E.H.P.} = \frac{D^{2/3}v^3 \times \text{P.C.}}{K_a}.$$

A moderate deviation in the first two requirements may not seriously affect the value of the method, but such is not the case with deviation from the third requirement.

Law of Comparison

1. **Corresponding Speeds.** The corresponding speeds for similar ships are proportional to the square roots of their lengths.

2. **Displacements.** Similar ships have displacements proportional to the cubes of their lengths.

3. **Corresponding Speeds.** The corresponding speeds for similar ships are proportional to the sixth roots of their displacements.

4. **Horse-powers.** The horse-powers of similar ships at corresponding speeds are proportional to the seven-sixths powers of their displacements.

This rule (4) is not strictly correct, however, as the frictional resistance does not follow the law of mechanical similitude.

5. **Variation of Power wi'h Speed.** Where the difference between the two speeds compared is small, we may assume that "The power for a ship is proportional to the cube of the speed," although this exponent may be widely departed from at high speeds.

6. **Variation of Power with Variation in Displacement.** For small changes in draught, we may assume that "The powers vary as the nth power of the Displacement," where n may vary from $\frac{2}{3}$ for large ships of moderate speed to $\frac{1}{6}$ for ships and boats of high speed.

In comparisons of hulls for similarity of form, Sheet 17 should always be employed on account of the great influence of $\frac{B}{L.L.W.L.}$ and coefficient of immersed midship section on fullness of lines.

Independent Estimate

The tow-rope resistance of a vessel is divided into three parts: surface or frictional resistance, residual resistance, and appendage resistance. The residual resistance is again divided into wave-making, eddy-making, and stream-line resistance.

The equation used for the calculation of frictional resistance is

$$R_f = fWv^n,$$

in which R_f is the force, in pounds, required to overcome the surface resistance, W is the wetted surface, in square feet, and v is the speed, in knots per hour. f and n are quantities taken from tables which can be obtained from any work on the "Resistance of Ships," and which are included here.

The equation used for finding the residual resistance is given as

$$R_w = \frac{bD^{2/3}v^4}{L}, \quad 4$$

where D, v, and L are the displacement in tons, the speed in knots per hour, and the length on the load water line, in feet. b is a numerical factor, having a value for long, fine ships of about .35; moderately fine ships, .40; ships broad in propor-

tion to length but with fine ends, .45; freighters, .5. The value of b is also likely to be affected by speed, especially when the speed-length ratio is high.

Total Bare Hull Resistance. As stated before, this is the sum of the two resistances, frictional and residual, and the equation for it is

$$R=R_f+R_w=fWv^n+\frac{bD^{2/3}v^4}{L}.$$

Using this equation in the estimation of the E.H.P., the equation for net E.H.P. takes the form

$$\text{E.H.P.}=0.00307\left(fWv^{n+1}+\frac{bD^{2/3}v^5}{L}\right),$$

where the various letters have the same significance as before.

Wetted Surface. This is determined from the lines of the ship and is a tedious operation. The surface is computed in square feet. For a preliminary design, the wetted surface may be computed by the equation $W=C\sqrt{DL}$, where D is the displacement, in tons, L the length on load water line, and C a coefficient depending on the beam and draught.

Model Experiments

The fourth method for determining power is by aid of model experiments in a towing basin. To illustrate the method, suppose that the tow-rope resistance for a paraffin model 20 ft. long is 12.8 lb., when towed at the speed corresponding to 25 knots for the full-sized vessel which has a length on load water line of 700 ft., then

$$v_m : 25 :: \sqrt{20} : \sqrt{700} \quad \therefore \quad v_m=4.23 \text{ knots.}$$

The wetted surface of the vessel is 67,540 sq. ft., therefore, the wetted surface of the model:

$$S_m : 67{,}540 :: 20^2 : 700^2 \quad \therefore \quad S_m=55.1 \text{ sq. ft.}$$

The friction factor and the exponent taken from Froude's tables are

$$f=0.00834 \text{ and } n=1.94;$$

therefore the frictional resistance is

$$=f \times S_m \times v_m^n = 0.00834 \times 55.1 \times 4.23^{1.94} = 7.54 \text{ lb.}$$

The total frictional resistance of the full-sized vessel, $f=0.00847$ and $n=1.825$, is $0.00847 \times 67{,}540 \times 25^{2.825}$ and the E.H.P. (frictional).

$$0.00307 \times 0.00847 \times 67{,}540 \times 25.^{2.825} = 15{,}600.$$

Taking the frictional resistance of the model from the total tow-rope resistance of the model, gives for the residual resistance

$$12.8 - 7.54 = 5.26 \text{ lb.}$$

The corresponding residual resistance for the ship is

$$R_W : 5.26 :: 700^3 : 20^3. \quad \therefore \quad R_W = 225{,}500 \text{ lb.}$$

At 25 knots the E.H.P. required to overcome this residual resistance will be $0.00307 \times 225{,}500 \times 25 = 17{,}310$.

The total E.H.P. will then be $15{,}600 + 17{,}310 = 32{,}910$.

In all the above methods, the results obtained are those for the bare hull only, and the appendage resistance increase called for by Sheet 18 must be applied before we are in a position to compute correctly the propeller and the indicated and shaft horse-powers.

(Credit must be given Peabody's work on "Propellers" for the major part of the above sections on "Resistance of Ships."—C. W. D.)

TABLE I

VALUES OF C FOR WETTED SURFACES

$B \div H$	C	$B \div H$	C	$B \div H$	C
2.0	15.63	2.5	15.50	3.0	15.62
2.1	15.58	2.6	15.51	3.1	15.66
2.2	15.54	2.7	15.53	3.2	15.71
2.3	15.51	2.8	15.55	3.3	15.77
2.4	15.50	2.9	15.58	3.4	15.83

B = beam. H = draught.

Table II

FROUDE'S SURFACE-FRICTION CONSTANTS

GIVEN BY TAYLOR

Surface-friction Constants for Paraffin Models in Fresh Water

Exponent $n = 1.94$

Length, Feet.	Coefficient.	Length, Feet.	Coefficient.	Length, Feet.	Coefficient.
2.0	0.01176	10.0	0.00937	14.0	0.00883
3.0	0.01123	10.5	0.00928	14.5	0.00877
4.0	0.01083	11.0	0.00920	15.0	0.00873
5.0	0.01050	11.5	0.00914	16.0	0.00864
6.0	0.01022	12.0	0.00908	17.0	0.00855
7.0	0.00997	12.5	0.00901	18.0	0.00847
8.0	0.00973	13.0	0.00895	19.0	0.00840
9.0	0.00953	13.5	0.00889	20.0	0.00834

Table III

SURFACE-FRICTION CONSTANTS FOR PAINTED SHIPS IN SEA WATER

Exponent $n = 1.825$

Length, Feet.	Coefficient.	Length, Feet.	Coefficient.	Length, Feet.	Coefficient.
8	0.01197	40	0.00981	180	0.00904
9	0.01177	45	0.00971	200	0.00904
10	0.01161	50	0.00963	250	0.00897
12	0.01131	60	0.00950	300	0.00892
14	0.01106	70	0.00940	350	0.00889
16	0.01086	80	0.00933	400	0.00886
18	0.01069	90	0.00928	450	0.00883
20	0.01055	100	0.00923	500	0.00880
25	0.01029	120	0.00916	550	0.00877
30	0.01010	140	0.00911	600	0.00874
35	0.00993	160	0.00907		

Estimate of Appendage Resistance: The resistance exerted by the appendages attached to the underwater body of a ship, that is, the resistances of the shaft struts, of the bilge and docking keels, etc., is generally assumed, and in the writer's opinion correctly so, to vary according to the Law of Comparison, and, on this assumption, when reduced fac-similes of these appendages

Table VI

SURFACE-FRICTION CONSTANTS. EXPONENT, 1.825
Given by Denny

Length, Feet.	Coefficient.	Length, Feet.	Coefficient.	Length, Feet.	Coefficient.
40	0.00996	260	0.00870	550	0.00853
60	0.00957	280	0.00868	600	0.00850
80	0.00933	300	0.00866	650	0.00848
100	0.00917	320	0.00864	700	0.00847
120	0.00905	340	0.00863	750	0.00846
140	0.00896	360	0.00862	800	0.00844
160	0.00889	380	0.00861	850	0.00842
180	0.00884	400	0.00860	900	0.00841
200	0.00879	420	0.00859	950	0.00840
220	0.00876	450	0.00858	1000	0.00839
240	0.00872	500	0.00855		

Table V

TIDEMAN'S SURFACE-FRICTION CONSTANTS
Derived from Froude's Experiments

Surface-friction Constants for Ships in Salt Water of 1.026 Density

Length of Ship in Feet.	Iron Bottom Clean and Well Painted		Copper or Zinc Sheathed. Sheathing Smooth and in Good Condition.		Copper or Zinc Sheathed. Sheathing Rough and in Bad Condition	
	f	n	f	n	f	n
10	0.01124	1.8530	0.01000	1.9175	0.01400	1.8700
20	0.01075	1.8490	0.00990	1.9000	0.01350	1.8610
30	0.01018	1.8440	0.00903	1.8650	0.01310	1.8530
40	0.00998	1.8397	0.00978	1.8400	0.01275	1.8470
50	0.00991	1.8357	0.00976	1.8300	0.01250	1.8430
100	0.00970	1.8290	0.00966	1.8270	0.01200	1.8430
150	0.00957	1.8290	0.00953	1.8270	0.01183	1.8430
200	0.00944	1.8290	0.00943	1.8270	0.01170	1.8430
250	0.00933	1.8290	0.00936	1.8270	0.01160	1.8430
300	0.00923	1.8290	0.00930	1.8270	0.01152	1.8430
350	0.00916	1.8290	0.00927	1.8270	0.01145	1.8430
400	0.00910	1.8290	0.00926	1.8270	0.01140	1.8430
450	0.00906	1.8290	0.00926	1.8270	0.01137	1.8430
500	0.00904	1.8290	0.00926	1.8270	0.01136	1.8430

are attached to the trial model of a vessel a curve of effective horse powers for the hull and appendages is obtained.

Unfortunately such a curve is not always furnished the designer, and when only the effective horse-power curve for the bare hull, or the estimate of the effective horse-power for the bare hull for any desired speed is supplied to the engineer, it becomes his task to correct this curve or estimate for the additional effective horse-power required by the appendages.

These appendages usually consist of the following, the various items being given in the order in which they most frequently occur:

1. Rudder and Stern Post.
2. Bilge Keels.
3. Struts, Bosses, and Shafting.
4. Docking Keels.
5. Small Scoops over openings in hull.
6. Large Scoops over openings in hull.

All other appendages that may be fitted are regarded as extraordinary and must be allowed for by the designer.

No. 1 is encountered in all vessels, either single or multiple screw.

No. 2 is met with in most vessels of any considerable size.

No. 3 exists only in vessels having two or more propellers, although in some cases of single-screw vessels, the dead wood may be cut away and the propeller shaft supported by a strut. In some cases of twin-screw ships, the form of stern known as the "Lundborg" stern may be used and there will be no struts. In such a case the appendage resistance will be less than when struts are fitted.

No. 4 is only met with in large, heavy vessels where such keels are required to better distribute the weight of the hull when docking.

No. 5 is found in all torpedo boats and destroyers built at the present date.

No. 6 is found in those torpedo boats and destroyers built from ten to twelve years ago.

The resistances due to the bilge and docking keels and shafts

are probably those due to their wetted surfaces only, and can be calculated as such. The other appendages enter into the total residual resistance (wave and eddy making), and are estimated as varying according to Froude's Law of Comparison.

Sheet 18. In preparing the curves of appendage resistance as given on Sheet 18, advantage has been taken of trials made in the Washington Model Tank by Naval Constructor (now Chief Constructor) D. W. Taylor, U. S. N., on models of battleships and destroyers, where, the model having been run through a series of speeds while fitted with appendages, these appendages were removed, one by one and other series were run after each removal, until the bare hull condition was reached when a final series of trials was made.

The reductions in resistances caused by refining the appendages were obtained from model tank trials of similar vessels but ones in which more care had been taken in placing appendages and in locating them so that their axes would more nearly coincide with the lines of flow of the water in proximity to the hull.

The curves as shown are cumulative and are erected on values of speed of ship (v) divided by the square root of the length on the load water line ($\sqrt{\text{L.L.W.L.}}$), as abscissas, the ordinates being percentages of the bare hull resistances of the vessel at these same abscissa values.

In Taylor's work on "The Speed and Power of Ships" is shown the following figure:

This figure shows the relations between speed of ship in knots, v, length of ship in feet, L, and values of $v \div \sqrt{L}$. The shaded areas indicate humps in the bare hull resistance curves while the clear areas between the shaded areas indicate hollows.

Returning to Sheet 18, and comparing it with Fig. 2, it is seen that the first hump in the bare hull curve extends from about $v \div \sqrt{L} = .75$ to $v \div \sqrt{L} = .83$, and that a corresponding hump in the appendage resistance curve attains its maximum value at $v \div \sqrt{L} = .75$. Fig. 2 shows another hump at $v \div \sqrt{L} = 1.0$ to 1.09 but the appendage curve shows no corresponding rise. Turning again to Fig. 2, a wide hump extending from $v \div \sqrt{L} = 1.25$ to $v \div \sqrt{L} = 1.65$ is found and on Sheet 18 is found a cor-

responding hump in the curves of appendage resistance. It will be noted that no matter which class of appendages is shown, the humps are in evidence. Evidently the causes producing the

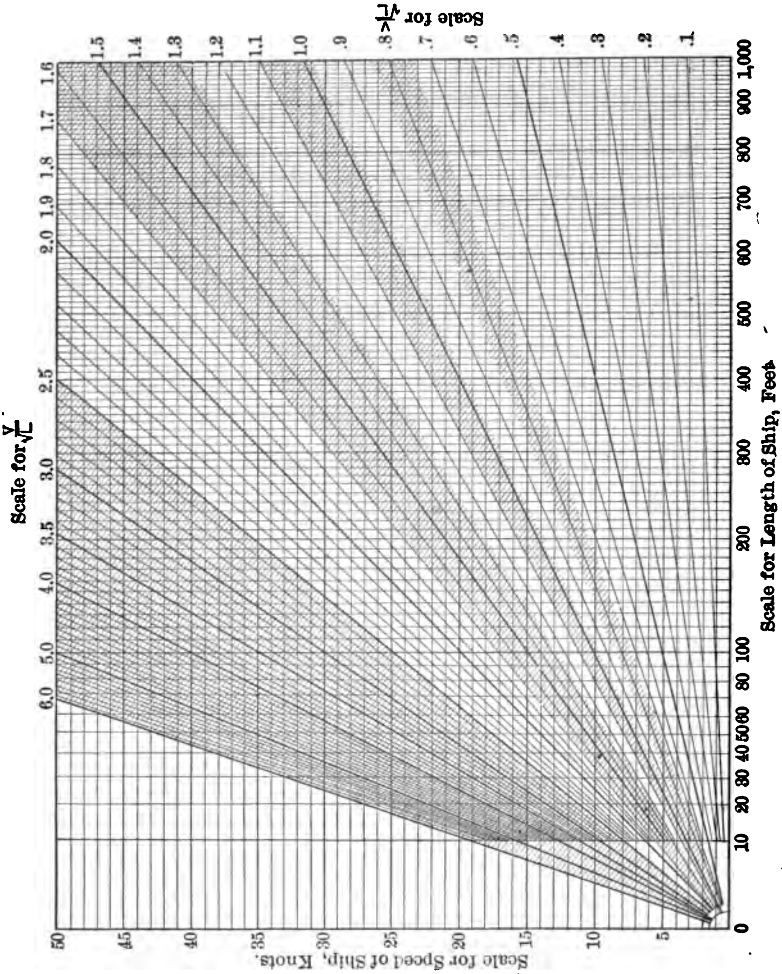


FIG. 2.—Diagram Indicating Humps and Hollows in Curves of Resistance.

humps in the resistance curve of the bare hull produce an augment of resistance to an even greater degree in the cases of the appendages.

Sheet 18 is built up as follows: The base of zero appendage resistance taken as the bare hull and rudder. The bilge keels

are then added and the appendage resistances for the various values of $v \div \sqrt{\text{L.L.W.L.}}$ rise to the values given by curved line marked 1.

Two propeller shafts with struts well aft, and with the strut section axes in the direction of motion of the vessel are then added, and the resistances rise to the curve numbered 2. This curve is a combination of two curves, one from $v \div \sqrt{\text{L.L.W.L.}} = 0$ to $= .95$ being taken from the model tank results for a large, heavy vessel while the portion from .95 to the end was obtained from the corresponding curves of a light, fast ship.

Again applying more appendages, docking keels were added in one case, and in the other the large shafts and struts of the two shaft arrangement were replaced by the four much smaller shafts and struts required to transmit the same total power as was transmitted by the two shafts. In the case of the light, fast vessel, while the two shaft arrangement was retained, injection scoops were added. The new percentage resistance curve is marked 3.

Now, returning to the two shaft arrangement, an additional strut was placed on each shaft, located well forward and with the axes of its sections made parallel to the stream lines of the water close to the hull. The appendage percentage resistance in this case rises from 3 to 4.

Removing these forward struts and fitting in their place others having their section axes parallel to the direction of motion of the vessel caused the appendage percentage resistances to rise from curve 3 to curve 5. All of these curves except two have been extended from $v \div \sqrt{\text{L.L.W.L.}} = .95$ to the extreme right hand of the sheet by maintaining approximately the same ratio between them and curve 2 as existed at $v \div \sqrt{\text{L.L.W.L.}} = .73$.

In actual service where the vessel is propelled by its own propellers, the resistances indicated by the humps are not in evidence. The humps are caused by abnormal increases in wake and these abnormal wakes deliver a large "wake gain" to the propellers, increasing the nominal propulsive efficiency of the hull and propeller by a considerable amount, in some cases to what may be regarded as almost unbelievable.

Finally it must be borne in mind that the appendage percentage resistance for any given vessel varies with the displacement of the vessel and that the only satisfactory manner of estimating this percentage is to tow the model at the displacement corresponding to the proposed trial displacement of the actual ship.

There is an additional curve, No. 6, shown on this same sheet which is the appendage resistance curve for a vessel of the merchant type, fan-tail stern, twin screw; the appendages are two struts, one per shaft, small bilge keels, rudder post and rudder.

CHAPTER III

EARLY INVESTIGATIONS FOR OBTAINING DATA FOR THE DESIGN OF SCREW PROPELLERS. EMPIRICAL FORMULAS

Numerous experiments were carried out during the period from 1843 to 1848, in the British naval vessels *Rattler*, *Dwarf* and *Minx*, and in the French naval vessel *Pelican*, to ascertain the effect produced by varying the characteristics of propellers.

The experiments in the *Rattler* commenced in 1843, and their main purpose was to ascertain the best length of propeller (fore and aft) to obtain a maximum speed of ship. The original propeller had a fore-and-aft length of 5 ft. 6 in., and this was successively reduced to 4 ft. 3 in., 3 ft., 1 ft. 6 in., and 1 ft. 3 in. An advantage was found to result from diminishing the length. Various kinds of propellers were tried including some with flat bands set at an angle with the axis, but it was found that the ordinary two-bladed screw with a uniform pitch was as efficient as any propeller of the different varieties tested.

The main purposes of the experiment which were made in the *Dwarf* in 1845, were to determine the proper pitch and length of the propeller relatively with its diameter. It was found that the speed of the vessel increased somewhat as the length of the propeller was diminished, but that relatively with the power consumed, the result obtained with the shortest propeller was worse than with the longest of them.

In 1847 and 1848, experiments in the *Minx* were made to determine the relative efficiencies of propellers with uniform and with variable pitches. Of the latter, propellers with axially increasing pitch, with radially expanding pitch increasing from the hub towards the circumference, and propellers in which the pitch increased both radially and axially were tried. The conclusion reached from these last series of experiments was that the

benefit obtained by departure from the form produced by a uniform pitch was found to be very inconsiderable, if any.

In all of the preceding cases the actual thrust of the propellers was measured by means of dynamometers fitted on the propeller shafts. In the series of tests which were made on the *Pelican*, while more elaborate and which appear to have been conducted with greater scientific accuracy than the British tests, no dynamometer was fitted.

The experiments conducted in the *Pelican* in 1847 and 1848 were repeated in 1849 on board the same vessel, using propellers of larger diameter than the original, and the results obtained in the earlier experiments were corroborated.

The object aimed at in the *Pelican* tests was the determination of the specific efficiency of all kinds of screw propellers in vessels of every size, proceeding at every speed, and under all circumstances of wind and sea, to the end that the particular species of propeller most proper for a given vessel might be readily specified. Another object in view was the determination of the value of the revolving force that it was necessary to bring to act upon the propeller shaft to obtain any definite number of revolutions in a given time, supposing, of course, that the form of the vessel was known as well as the dimensions of the propeller. It is readily seen that the problem thus proposed for solution is the general problem of screw propulsion whose correct answer has been sought by many since the early days of the *Pelican* tests.

The conclusions arrived at by these tests may be briefly summarized, as follows: " Not only does the efficiency of a screw increase with its diameter, or rather with the relative resistance, but the proper ratio of the pitch to the diameter, and the corresponding fractions of the pitch, vary with the relative resistance, the ratio of the pitch to the diameter diminishing when the fraction of the pitch increases, while the fraction of the pitch varies with an inverse progression."

Bourne, " Treatise on Screw Propellers," states that these tests enable us, with any given diameter, to specify the best pitch and the best length of screw that can be employed, whether

the screw is formed with two, four, or six blades. For taking K as the resistance per square metre of immersed midship section (equal 6 kilogrammes or 13.23 lb. per square metre at a speed of 1 metre per second), B^2 the area of immersed midship section in square metres, D the diameter of the screw in metres, and P the pitch of the screw in metres, then

$$D=\sqrt{\frac{B^2}{K}},$$

and D multiplied by the ratio of pitch to diameter, given in an empirical table obtained by the experiments, will give P.

Finally, $\dfrac{P\times \text{ fraction of Pitch (tabulated)}}{\text{number of Blades}}=$ Length of screw.

For years after these experiments had been completed there was apparently no systematic attack made upon the propeller problem, engineers being apparently perfectly well satisfied with the results obtained by the use of such formulas as the following:

d is the diameter of the L.P. cylinder of the engine in feet;
L is the stroke in feet;
P_c is the block coefficient of vessel;
Z is a multiplier $=(2.4-P_c)$ for twin screws;
and $=(2.7-P_c)$ for single screws;
R = revolutions per minute.

Rule I. D = diam. of screw in feet $=Z\times\sqrt{d\times L}$,

Rule II. D = diam. of screw in feet $=x\times P_c\sqrt[3]{\dfrac{\text{I.H.P.}}{R}}$,

in which for single screw, $x=7.25$
for twin screw, $x=6.55$
for quadruple screw, $x=6.25$
for turbine-driven center screw, $x=6.55$
for turbine-driven wing screw, $x=5.75$
for ocean express steamer, $x=7.61$
for ocean express steamer, $x=6.88$
for ocean express steamer, $x=6.51$
for ocean express steamer, $x=6.88$
for ocean express steamer, $x=6.04$

In no case must P_c have a less value than .55.

Rule III. $A=\pi D^2\div 4$, where D = diameter of propeller.
The thrust of the propeller in pounds $=2A\times V(V-v)$.
The work done per minute $=2A\times V(V-v)\times 60v$ ft. lbs.
$V=P\times R$, and v = speed of ship in feet per minute.

$$\text{The thrust horse-power}=\frac{2A\times V(V-v)\times 60v}{33{,}000}=\frac{A\times V(V-v)\times v}{275}.$$

If E is the efficiency of propeller and engine,

$$\text{I.H.P.}=\frac{A\times V(V-v)\times v}{275\ E}.$$

Let $(V-v)\div V=s$ = slip in per cent, then
$V-v=sV$
and $v=V-sV$, or $V(1-s)$.

Substituting these values in the equation for I.H.P., there results:

$$\text{I.H.P.}=.7854\frac{D^2\times V\times sV\times V}{275\ E}(1-s)=\frac{D^2\times V^3(1-s)s}{350\ E}.$$

But $V=P\times R$; therefore

$$\text{I.H.P.}=\frac{D^2\times(P\times R)^3(s-s^2)}{350\ E}.$$

In actual practice there are disturbing causes which increase the value of the factor above 350, as with very large hubs the column of water flowing through the propeller is hollow, and the equivalent diameter is then less than D. Also the apparent slip s is less than the real slip. To know how large a real propeller should be for actual practice another factor is necessary, hence

Rule IV. For good work and high efficiency:

$$D=\sqrt{\frac{\text{I.H.P.}}{x(1-x)}\times\left(\frac{\text{C}}{P\times R}\right)^3}=\sqrt{\frac{\text{I.H.P.}}{x-x^2}\times\left(\frac{\text{C}}{P\times R}\right)^3},$$

where $x=.2P_c-s$, where P_c is the block coefficient of the vessel and s the apparent slip.

For single-screw ships and for center propellers of triple-screw ships:

$x=.18P_c+s$, and $C=450$.

To determine the pitch of the propeller, using the same notation as in the rules for diameter, calling the area of the propeller disc A, the diameter D, and the pitch P,

Rule V. $A = .7854D^2$.

Thrust $= 2A \times V(V-v)$ lb. $= 1.57D^2 \times V(V-v)$.

Taking $V(V-v) = sV$, where $s =$ the apparent slip,

Thrust $= 1.57D^2 \times xV^2$ lb.

Assuming that for any given speed and size of ship, the thrust remains constant, then

$sD^2V^2 =$ constant, that is

$D \times V$ varies inversely as $\sqrt{s}$.

Let v be the speed of the ship in knots, V the speed of the propeller $= v \div (1-s)$, then

$$\frac{P \times R}{101.33} = \frac{v}{1-s} \quad \therefore \quad P = \frac{101.33\, v}{R(1-s)}.$$

In all of these rules the only controlling influences that are considered are the power of the engine driving the propeller, the desired revolutions, and the actual block coefficient of the vessel. No attention is paid to the variation in block coefficient that is produced by varying the length of the middle body of the vessel, the fore and the after bodies remaining constant; nor is any attention paid to the variations of the speed of wake of the vessel at different positions in it, these variations modifying the percentage apparent slip that should be used with any particular set of after body lines.

The empirical rules for determining the developed area of the propeller are equally as crude as those for obtaining pitch and diameter, as the following will show:

Rule VI. Area of developed surface in square feet $= K\sqrt{\dfrac{\text{I.H.P.}}{R}}$

where $K = P_c \times M$ and

for four-bladed single screws, $M = 20$; for twin, 15;
for three-bladed single screws, $M = 19$; for twin, 14.3;
for two-bladed single screws, $M = 17.5$; for twin, 13.1.

Rule VII. Calling the developed surface, A_s; P_r the pitch ratio $= P \div D$; D the diameter, in feet; V the velocity of the

propeller in feet per second $=(P\times R)\div 60$; and G a coefficient which varies in value from .42 for long narrow blades to .5 for broad and short turbine propeller blades, then

$$T = \text{thrust in pounds} = (D\times\sqrt{A_s}\times V^2\times G)\div P_r.$$

From this formula for thrust, the following formula for developed surface is obtained:

$$A_s = \text{Area of developed surface} = \left(\frac{T\times P_r}{D\times V^2\times G}\right)^2.$$

Taking the apparent slip as a percentage of V, so that it is represented by sV, then

$$\text{Speed of ship} = v = V - sV = V(1-s).$$

The efficiency of the engine and propeller being represented by E, then

$$T = (\text{I.H.P.}\times 33{,}000\times E)\div 60v = (\text{I.H.P.}\times 550\times E)\div V(1-s).$$

Substituting this value of T in the first equation for A_s, there results,

$$A_s = \left\{\frac{\text{I.H.P.}\times 550\times E}{V(1-s)}\times\frac{P_r}{D\times V^2\times G}\right\}^2 = \left\{\frac{\text{I.H.P.}\times 550\times E\times P_r}{D\times V^3(1-s)\times G}\right\}^2.$$

The values usually assumed for 550 E are given as follows:

For ordinary merchant cargo steamers,	550 $E=330$
For express and naval reciprocators,	550 $E=360$
For turbine-driven ships,	550 $E=38$

Having now shown some of the purely empirical formulas formerly generally and at present, occasionally used in the determination of propeller dimensions, it is time to turn to the other extreme and examine the work of the pure theorists, and this will be taken up in the following chapter,

CHAPTER IV

THEORETICAL TREATMENT OF SCREW PROPELLER

The Screw Propeller

Theories of Design

The three most important theories of design, given in the order of their importance, are: 1. Froude's; 2 Rankine's, and 3. Greenhill's. The assumptions for each of these being as follows:

Froude. Assumes the element as a small plane moving through the water along a line which makes a small angle with the direction of the plane. He then takes the normal pressure upon the elementary area, which gives propulsive effect to vary as the area, as the square of its speed and as the sine of the slip angle.

Rankine. The fundamental assumption is that as the propeller advances with a certain slip, all the water in an elementary ring of radius r is given a certain velocity in a direction perpendicular to the face of the blade at that radius. Then, from the principle of momentum, the thrust from the elementary ring is proportional to the quantity of water acted upon in one second, and to the sternward velocity communicated to it.

Greenhill. Approaches the problem from a direction entirely different from that of either of the two preceding theorists. He assumes that the propeller is working in a fixed tube with closed end. The result is that the motion transmitted to the water is wholly transverse. The blade is assumed perfectly smooth, so that the pressure produced by the reaction of the water is normal to the blade and has a fore-and-aft component which produces thrust.

In all the above theories, the loss by friction is taken as that

due to the friction of the propelling plane moving edgewise, or nearly so, through the water.

In all the theories in connection with which mathematical methods are to be used, it is practically necessary to regard the blade as having no thickness. This is a serious defect in the theories, as they all use a true slip based upon true pitch and consider the designed pitch of the driving surface of the blade as this true pitch. The fact of the matter is that the face pitch of a blade with thickness, or its nominal pitch as it may be called, is very different from the true or actual pitch, and this fact causes complications in using the mathematical formulas.

It would be necessary, in case these theoretical formulas were adhered to, to compare each formula with experimental results and select that one which seemed to agree more closely. Then, using this as a semi-empirical formula, with coefficients and constants deduced from experiments or experience, problems could be satisfactorily dealt with. When the vast number of various conditions for which we may be called upon to design a propeller are considered, it is readily seen how impossible it would be to tabulate the correctors which would be required to cover all, or even a large number of these conditions.

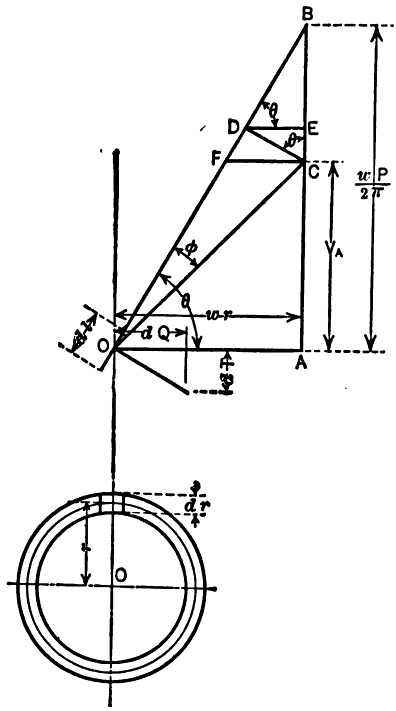


FIG. 3.

In order to give a thorough understanding of the study that has been put on the subject of the propeller, it will be well to present these in their mathematical form for the determination of thrust and torque, and in doing this

nothing better can be done than to give Naval Constructor Taylor's presentation of the theories, as set forth in his work on "The Speed and Power of Ships."

Fig. 3 indicates the motion of a small elementary plane blade area of radius r, breadth dr in a radial direction, and circumferential length dl. This element is seen with its center at O.

If w is the angular velocity of rotation of the shaft, the circular velocity of the element is wr. AOB is the pitch angle θ, BC the slip and BOC the slip angle ϕ. Now, $\tan\theta = P \div 2\pi r$. Considering Fig. 3 as a diagram of instantaneous velocities, the line OA or wr represents the circular velocity of the element. If there were no slip, the actual velocity along the helical path would be OB and AB would represent the axial velocity or the velocity of advance, and

$$AB = OA \tan\theta = wr \tan\theta = wr\frac{p}{2\pi r} = \frac{wp}{2\pi}.$$

When there is slip the circular velocity of the element is unchanged, but the velocity of advance becomes AC, the speed of the screw is the same as the speed of advance when the slip is zero.

Denote the percentage slip by s, then

$$s = BC \div AB = (AB - AC) \div AB = \left(\frac{wp}{2\pi} - V_A\right) \div \frac{wp}{2\pi} = 1 - V_A\frac{2\pi}{wp}.$$

From which the speed of advance

$$V_A = \frac{wp}{2\pi}(1 - s) \text{ and } BC = s\,\frac{wp}{2\pi}.$$

If w is taken as the angular velocity per second and r is taken in feet, then OA, or the circular velocity, is in feet per second and therefore all other velocities will be in the same units.

Finally, taking the components, we have:

Velocity of element in direction perpendicular to its plane

$$= CD = BC \cos\theta = s\frac{wp}{2\pi}\cos\theta.$$

Axial or rearward component of this velocity

$$=CE=CD\cos\theta=s\frac{wp}{2\pi}\cos^2\theta.$$

Transverse component of the same velocity

$$=DE=CD\sin\theta=s\frac{wp}{2\pi}\sin\theta\cos\theta.$$

Rankine's Theory. Referring to Fig. 3, and considering the annular ring of mean radius r.

Annular area $=2\pi rdr$.

Volume of water acted on per second

$$=2\pi rdr\times AE=2\pi rdr\times\frac{pR}{60}(1-s\sin^2\theta).$$

Sternward velocity communicated

$$=EC=s\frac{wp}{2\pi}\cos^2\theta=s\frac{pR}{60}\cos^2\theta.$$

Therefore elementary thrust = mass of water per second $\times$ sternward velocity imparted $=dT=\frac{w}{g}\,2\pi rdr\,\frac{pR}{60}(1-s\sin^2\theta)\times s\frac{pR}{60}\cos^2\theta$

$$=\frac{w}{g}\,\frac{p^2R^2}{3600}s(1-s\sin^2\theta)\cos^2\theta\;2\pi rdr.$$

Let $q=\cot\theta=\frac{2\pi r}{p}$, then $2\pi rdr=\frac{p^2q}{2\pi}dq$; $\sin^2\theta=\frac{1}{1+q^2}$;

$$\cos^2\theta=\frac{q^2}{1+q^2}.$$

Whence

$$dT=\frac{w}{g}\,\frac{p^2R^2}{3600}s\left(\frac{q^2}{1+q^2}-s\frac{q^2}{(1+q^2)^2}\right)\frac{p^2}{2\pi}qdq$$

$$=\frac{w}{g}\,\frac{p^2R^2}{3600}\,\frac{p^2}{2\pi}s\left\{qdq-\frac{qdq}{1+q^2}-s\left(\frac{qdq}{1+q^2}-\frac{qdq}{(1+q^2)^2}\right)\right\}.$$

At the axis $q=0$. Neglecting the hub, if q denote now the co-tangent of the pitch angle of the blade tips, on integrating the expression for dT, is obtained.

$$T=\frac{w}{g}\frac{p^2R^2}{3600}\frac{p^2}{2\pi}s\left[\frac{q^2}{2}-\frac{\log_e(1+q^2)}{2}-s\left(\frac{\log_e(1+q^2)}{2}-\frac{1}{2}\frac{q^2}{1+q^2}\right)\right]$$

$$=\frac{w}{g}\frac{p^2R^2}{3600}\frac{p^2q^2}{4\pi}s\left[1-\frac{\log_e(1+q^2)}{q^2}-s\left(\frac{\log_e(1+q^2)}{q^2}-\frac{1}{1+q^2}\right)\right].$$

Now, $pq=2\pi r$; $p^2q^2=4\pi^2r^2$; $\frac{p^2q^2}{4\pi}=\pi r^2=\frac{\pi d^2}{4}$ if d is extreme diameter. Whence

$$T=\frac{w}{g}\frac{p^2R^2}{3600}\frac{\pi d^2}{4}s\left[1-\frac{\log_e(1+q^2)}{q^2}-s\left(\frac{\log_e(1+q^2)}{q^2}-\frac{1}{1+q^2}\right)\right],$$

and finally,

$$T=\frac{\pi w}{14400g}p^2d^2R^2s\left[1-\frac{\log_e(1+q^2)}{q^2}-s\left(\frac{\log_e(1+q^2)}{q^2}-\frac{1}{1+q^2}\right)\right],$$

and the torque,

$$Q=\frac{pT}{2\pi}.$$

W. Froude's Theory. If l is the total fore-and-aft blade length of all blades at radius r, then the total elementary plane area at this radius is ldr. This area advances at the angle ϕ (Fig. 3), with velocity OC, and from Froude's experiments if a is a thrust coefficient, the resultant pressure normal to the blade is

$$=ldr\ a\ \overline{OC}^2\sin\phi.$$

The elementary thrust is equal to this pressure $\times\cos\theta$. Then

$$dT=ldr\ a\ \overline{OC}^2\sin\phi\cos\theta.$$

Now

$$\overline{OC}^2=w^2r^2\frac{w^2p^2}{4\pi^2}(1-s)^2=\frac{w^2p^2q^2}{4\pi^2}+\frac{w^2p^2}{4\pi^2}(1-s)^2$$

$$=\frac{p^2R^2}{3600}\{q^2+(1-s)^2\}$$

$$\sin\phi=\frac{CD}{OC}=\frac{s\frac{wp}{2\pi}\cos\theta}{\frac{wp}{2\pi}\sqrt{q^2+(1-s)^2}}=\frac{s}{\sqrt{q^2+(1-s)^2}}\cos\theta.$$

Also

$$\cos^2\theta=\frac{q^2}{1+q^2}.$$

Whence

$$dT=ladrs\frac{p^2R^2}{3600}\frac{q^2}{1+q^2}\sqrt{q^2+(1-s)^2}$$

$$=\frac{a}{3600}p^2R^2d^2\frac{l}{d}\frac{q^2}{1+q^2}\sqrt{q^2+(1-s)^2}\frac{dr}{d}$$

$$=\frac{a}{3600}p^2d^2R^2\frac{l}{d}\frac{p}{d}\frac{q^2}{1+q^2}\sqrt{q^2+(1-s)^2}\frac{dq}{2\pi}.$$

Whence, deducting the hub,

$$T=\frac{a}{3600}p^2R^2ds\int_0^{q}\frac{l}{d}\frac{q^2}{1+q^2}\sqrt{q^2+(1-s)^2}\frac{dq}{2\pi}.$$

The quantity under the integral sign is dependent only on shape and proportions of the propeller and independent of its dimensions. Let it be denoted by the symbol X. Then

$$T=\frac{a}{3600}p^2R^2dsX, \text{ and}$$

$$Q=\frac{pT}{2\pi}.$$

Greenhill's Theory. Referring again to Fig. 3,

Elementary area $=2\pi rdr$.

Velocity of feed of the water $=AC=\frac{wp}{2\pi}(1-s)=\frac{pR}{60}(1-s)$.

Circular velocity $=s\frac{wp}{2\pi}\cot\theta=swr=s\frac{2\pi R}{60}r$.

Circular momentum per second.

$$=\frac{w}{g}2\pi rdr\frac{pR}{60}(1-s)s\frac{2\pi R}{60}r.$$

$$=\frac{w}{g}p\frac{R^2}{3600}s(1-s)4\pi^2r^2dr.$$

Torque = circular momentum $\times r$.

Whence $dQ = \frac{w}{g} p \frac{R^2}{3600} s(1-s) 4\pi^2 r^3 dr.$

$$dT = \frac{2\pi dQ}{p} = \frac{w}{g} \frac{R^2}{3600} s(1-s) 8\pi^3 r^3 dr.$$

Integrating from $r = 0$ to $r = \frac{d}{2}$, there results

$$T = \frac{w}{g} \frac{R^2}{3600} s(1-s) 2\pi^3 r^4 = \frac{3w}{28800g} d^4 R^2 s(1-s).$$

And
$$Q = \frac{pT}{2\pi}.$$

The equations for thrust and torque are further modified, in all the theories, by corrections for frictional and head resistances, the thrust being decreased and the torque increased.

The decrease from thrust for friction

$$= T_f = f p^3 d R^2 \int \frac{l\sqrt{1+q^2}}{2\pi d} dq = f p^3 d R^2 Y,$$

where
$$Y = \int \frac{l\sqrt{1+q^2}}{2\pi d} dq.$$

The addition to the torque for friction

$$= Q_f = \frac{p}{2\pi} f p^3 d R^2 \int \frac{q^2 l \sqrt{1+q^2}}{2\pi d} dp$$

$$= \frac{p}{2\pi} f p^3 d R^2 Z, \text{ where } Z = \int \frac{l}{2\pi d} q^2 \sqrt{1+q^2}\, dp.$$

In both equations f denotes the coefficient of friction and is taken sufficiently large to cover all edgewise resistance, both skin and head resistance together.

Final Formulas for the Three Theories

The resulting equations for thrust and torque for the three foregoing theories may be expressed thus:

Rankine:

$$T=p^2d^2R^2(\alpha s-\beta s^2)-fdp^3R^2Y.$$

$$Q=\frac{p}{2\pi}[p^2d^2R^2(\gamma s-\delta s^2)+fdp^3R^2Z].$$

Froude:

$$T=p^3dR^2(\alpha s-\beta s^2)-fdp^3R^2Y.$$

$$Q=\frac{p}{2\pi}[p^3dR^2(\gamma s-\delta s^2)+fdp^3R^2Z].$$

Greenhill:

$$T=d^4R^2(\alpha s-\beta s^2)-fdp^3R^2y.$$

$$Q=\frac{p}{2\pi}[d^4R^2(\gamma s-\delta s^2)+fdp^2R^3Z].$$

These equations simply show the form of the expressions, and do not imply that the values of α, β, γ and δ are the same in all the theories, but simply imply that in each case the values of these factors will be constant for a given propeller. The actual values of these factors will vary with the theory used.

Having obtained the values of T and of Q, the efficiency can be obtained as follows: Denoting the pitch by p, as before, the revolutions per minute by R, and the slip by s, the speed of advance of the propeller is $p(1-s)R$, and the useful work done per minute is $Tp(1-s)R$, while the gross work delivered to the propeller is $Q\times 2\pi R$.

$\therefore$ Efficiency = (Useful Work) $\div$ (Gross Work) =

$$Tp(1-s)R\div 2Q\pi R=\frac{T}{Q}\,\frac{p(1-s)}{2\pi}=e.$$

CHAPTER V

PRACTICAL METHODS OF DESIGN. DESIGN BY COMPARISON. TAYLOR'S METHOD. BARNABY'S METHOD

THE practical methods of design can be divided into

1. By "Direct Comparison," when all the conditions for a satisfactory vessel of a similar form to the one under consideration are known.

2. By methods based on trials of model propellers in model tanks. This may also be classed under the head of "direct comparison," as the Laws of Comparison are assumed to cover propellers as well as hulls.

3. By methods based on actual trials of full-sized propellers in service over carefully measured courses.

The first method practically insures a propeller of equal propulsive efficiency with that of the propeller on the compared vessel, but gives no opportunity for improvement in performance.

The second method is open to the decided objection that the conditions under which the model screw is tried in the tank are radically different from those under which the full-sized screw operates. In fact, propellers whose models have shown high tank efficiencies have failed most signally in service, while other propellers whose models gave poor efficiency have delivered a high propulsive coefficient. This latter has been ascribed to a high hull efficiency but this explanation does not exactly satisfy when the fact is considered that where two or more propellers for the same vessel have been tested, that propeller whose model gave the highest efficiency has failed, while the propeller with the lower tank efficiency has succeeded.

The writer is inclined to the belief that the true cause of these discrepancies exists in the use of an incorrect method of

derivation of the model screw dimensions from those of the full-size propeller.

Method of "Design by Comparison"

The following orthodox method is taken from Peabody's "Naval Architecture," and is that which is generally used:

D_1 = Diameter of original propeller;
D_2 = Diameter of 2d propeller or of model propeller;
L_1 = Length of compared vessel;
L_2 = Length of new vessel or model hull behind which model screw should operate if fitted to a hull (which is not usually done);
P_1 = Pitch of original propeller;
P_2 = Pitch of 2d or model propeller;
R_1 = Revolutions of original propeller;
R_2 = Revolutions of 2d or model propeller;
v_1 = Speed of compared vessel;
v_2 = Corresponding speed of new or model hull.

Then
$$D_2 = D_1 \times \frac{L_2}{L_1} = D_1 r;$$
$$R_2 = R_1 \left(\frac{L_1}{L_2}\right)^{\frac{1}{2}} = \frac{R_1}{r^{\frac{1}{2}}};$$
$$v_2 = v_1 \left(\frac{L_2}{L_1}\right)^{\frac{1}{2}} = v_1 r^{\frac{1}{2}};$$

$P_2 = pP_1 = rP_1$ where same ratio of $\frac{P}{D}$ is retained for the model as that of the original propeller.

Apparent slip$_1$
$$s_1 = \frac{P_1 \times R_1 - 101.33 v_1}{P_1 \times R_1}.$$

Apparent slip$_2$,
$$s_2 = \frac{pP_1 \times \frac{R_1}{r^{\frac{1}{2}}} - 101.33\ v_1 r^{\frac{1}{2}}}{pP_1 \times \frac{R_1}{r^{\frac{1}{2}}}} = \frac{pP_1 \times R_1 - 101.33\ v_1 r}{pP_1 \times R_1};$$

which, when $p=r$ becomes $\frac{P_1 \times R_1 - 101.33\, v_1}{P_1 \times R_1}$, or the apparent slip of the model, is equal to the apparent slip of the original propeller.

Tip speed$_1 = R_1 \times \pi D_1$;

Tip speed$_2 = R_2 \times \pi D_2 = \frac{R_1}{r^{1/2}} \times \pi D_1 r = R_1 \pi D_1 r^{1/2}$; and

T.S.$_2 \gtrless$ T.S.$_1$, depending upon the value of r.

Again,

$$\text{I.H.P.}_2 = \text{I.H.P.}_1 r^{7/2};$$

$$\text{Disc area}_2 = \tfrac{1}{4}\pi D_1^2 r^2;$$

$$\text{Disc area}_1 = \tfrac{1}{4}\pi D_1^2;$$

$$P_2 \times R_2 = \frac{pP_1 \times R_1}{r^{1/2}};$$

I.T. per square inch disc area$_2$

$$= \frac{\text{I.H.P.}_1 \times r^{7/2} \times 33{,}000}{\frac{pP_1 \times R_1}{r^{1/2}} \times \frac{\pi}{4} D_1^2 r^2} = \frac{\text{I.H.P.}_1 \times 132{,}000 \times r^2}{pP_1 \times R_1 \times \pi D_1^2};$$

I.T. per square inch disc area$_1$

$$= \frac{\text{I.H.P.}_1 \times 33{,}000}{P_1 \times R_1 \times \frac{\pi}{4} D_1^2} = \frac{\text{I.H.P.}_1 \times 132{,}000}{P_1 \times R_1 \times \pi D_1^2},$$

$$\therefore\ \frac{\text{I.T.}_2}{\text{I.T.}_1} = \frac{r^2}{p}, \text{ or where } p=r,\ \frac{\text{I.T.}_2}{\text{I.T.}_1} = r.$$

In other words, with different percentage losses from blade friction due to change in tip speeds, the model screw is supposed to deliver an equal percentage of the power driving it as effective thrust, with the original propeller, and its apparent slip is supposed to be equal to that of the original propeller, although the thrusts per square inch of disc area have been changed in the ratio r, the two screws working under approximately the same conditions of resistance.

Taylor's Method of Design

This method of design is based entirely upon the trials of model propellers in the Model Tank, and from the results obtained were derived practical coefficients and constants for full-sized propellers.

The factors dealt with in this method of design are efficiency, diameter, pitch ratio, mean width ratio of the blade and blade thickness fraction.

In this method there is a primary variable, ρ fixed by the conditions of the problem. Its value is expressed by

$$\rho = R\sqrt{\frac{\text{S.H.P.}}{V_A^5}}.$$

where S.H.P. is the shaft horse-power absorbed by a propeller of D feet diameter at R revolutions per minute when advancing at a speed of V_A knots.

Another factor δ is expressed by the following equation:

$$\delta = D\frac{R^{2/3}}{(V_A \times \text{S.H.P.})^{1/6}}.$$

Diagrams of $p\delta$, efficiency and real slip for various pitch ratios, mean blade width ratios, blade thickness fractions and speed of wake for elliptical three-bladed propellers are prepared from model tank trials of model propellers, and from these the necessary factors for use are obtained.

In the above equations V_A is not the speed of the ship through the water but is the speed of advance of the propeller through the disturbed water in which it works.

For determining the thickness of the blades, Taylor has obtained the following expressions:

The compressive stress in pounds per square inch for blades of the usual ogival section

$$= S_c = 14C\frac{P_1}{R}\frac{1}{lt^2},$$

where C is a coefficient depending on radius and pitch ratio, P_1 is the shaft horse-power absorbed by the blade, R = the revolu-

tions per minute of the propeller and l and t are the width and thickness of the blade in inches. In determining S_c the values of C, l and t at about .2 the radius of the propeller should be used, this fraction of the radius being the approximate radius of the hub for three-bladed propellers of the built-up type, and also being approximately the point of maximum stress.

Put $l=12\,chd$, where d is diameter of propeller in feet, h is the mean width ratio of a blade and c is a coefficient depending upon the shape of the blade.

For the thickness t, calling the axial thickness of the blade Td, and the thickness at the tip kTd, then at .2 radius

$$t=12Td[k+.8(1-k)]=12Td(.8+.2k).$$

In practice k is seldom less than .1 or greater than .2. When $k=0$, $t=9.6Td$; $k=.1$, $t=9.84Td$; $k=.2$, $t=10.08Td$; hence it is a sufficient approximation to assume $t=10Td$.

Substituting these in the stress formula,

$$S_c=14C\frac{P_1}{R}\times\frac{1}{12chd}\times\frac{1}{100T^2d^2}=\frac{14C}{1200}\times\frac{P_1}{Rd^3}\times\frac{1}{chT^2}.$$

Let $C_1=\dfrac{14C}{1200}$, then

$$S_c=\frac{C_1P_1}{Rd^3}\times\frac{1}{chT^2}.$$

Let $\dfrac{C_1P_1}{Rd^3}=x$, $chT=y$, then

$$S_c=\frac{x}{y}.$$

Values of S_c are given as curves plotted on values of x and y.

Barnaby's Method

Mr. Sydney W. Barnaby has recast the results obtained by Mr. R. E. Froude from trials of model propellers into the following form for use in the designing of propellers:

He has chosen a standard wake value of 10 per cent, a coefficient of propulsion of .5, the resistance of the bare hull only

being considered, and a blade having an elliptical form of developed area, the major axis being the radius of the propeller and the minor axis being .4 that radius. The total developed area of the blade being the area of this ellipse less the area included within the radius of the propeller hub.

Barnaby's factors are tabulated, and as so tabulated are for four-bladed propellers, but can be used for three- or two-bladed propellers by taking account of Froude's determinations of the relative efficiencies of these numbers of blades.

The size and revolutions of the propeller are given by the following expressions:

$$A = \text{Disc Area} = C_A \times \frac{\text{I.H.P.}}{V^3},$$

$$R = \text{Revolutions} = C_R \times \frac{V}{D},$$

where $\quad D = \text{diameter of propeller in feet} = \sqrt{\frac{4A}{\pi}}.$

$\quad V = \text{Speed of ship in knots per-hour.}$

As the values of C_A and C_R vary with the pitch ratio, call this ratio

$$p = P \div D,$$

where $P =$ the pitch of the propeller in feet.

Then

$$A = C_A \frac{\text{I.H.P.}}{V^3}, \quad \ldots \ldots \ldots \quad (1)$$

$$\therefore \; C_A = \frac{AV^3}{\text{I.H.P.}},$$

$$R = C_R \frac{V}{D}, \quad \ldots \ldots \ldots \quad (2)$$

$$\therefore \; C_R = \frac{RD}{V},$$

$$k = \frac{\text{I.H.P.} \times R^2}{V^5}. \quad \ldots \ldots \ldots \quad (3)$$

Equation (3) is used as an aid in proportioning propellers which must have a given speed of revolutions.

For three-bladed propellers the formula for C_A becomes

$$C_A = \frac{.865 A V^3}{\text{I.H.P.}},$$

while for two-bladed propellers it becomes

$$C_A = \frac{.65 A V^3}{\text{I.H.P.}}.$$

Corrections for Variation in Wake, Estimated Propulsive Coefficient and in Blade Width Ratio

"The standard wake has been taken as 10 per cent of the speed of the vessel. In a very full ship it might be as much as 30 per cent. Therefore the speed of the ship, V, should be reduced when using the constants, by about 20 per cent for a very full ship, and by amounts varying from 20 per cent to nothing, as the fullness of form varies from "very full" down to what may be considered a "fairly fine" vessel when no correction may be made."

"The standard value of the propulsive coefficient $= \frac{\text{E.H.P.}}{\text{I.H.P.}}$, has been taken as .5. A correction can be made for any expected deviation from this assumed value. If the propulsive coefficient is estimated at 55 per cent, then the I.H.P. must be multiplied by $\frac{55}{50}$.

To correct for varying width ratios of blades, Professor C. E. Peabody suggests that the method proposed by Naval Constructor Taylor, be used, namely, to make the thrust proportional to the width of the blade.

Suppose the blade is .6 as wide as the radius of the propeller, then

$$A = C_A \times \frac{\frac{4}{6}\text{I.H.P.}}{V^3}. \quad \therefore \ C_A = \frac{3 A V^3}{2\text{I.H.P.}},$$

$$R = C_R \times \frac{V}{D}, \qquad \therefore \ C_R = \frac{RD}{V}.$$

Correction for Varying Values of Developed Area Ratio From the Standard

By assuming that the total thrust that can be delivered by any propeller of fixed pitch, diameter and revolutions will vary directly as the developed area ratio, a series of curves can be laid down as shown on Sheet 16, by which the values of

$$C_A = \frac{A V^3}{\text{I.H.P.}} = \frac{.92 A V^3}{\text{S.H.P.}}$$

and

$$C_R = \frac{RD}{V}.$$

can be obtained for any desired value of developed area, H.A., divided by disc area, D.A.

The values of C_A are shown as ordinates on the left of the sheet, the abscissa values being increasing values of H.A. ÷ D.A.

On the right are curves of pitch ratio, $\frac{P}{D}$, inclined close to the vertical, while the curves approximating more closely to the horizontal are those of propeller efficiency, not propulsive efficiency. These curves of $P \div D$ and of efficiency are both erected on values of $C_R = \frac{RD}{V}$ as abscissas.

In this equation, however, V does not equal the speed of the ship as in the Barnaby formula but equals the speed of the ship $\times$ a coefficient M, whose values change with the wake, and which must be obtained from the analysis, by means of these curves, of the trials of numerous vessels.

To obtain the correct value of M from the actual trial results of vessels, a value of $M = 1$ is first assumed, and with the I.H.P.; the V = speed of ship and the revolutions obtained on trial for this I.H.P. and V, together with the actual diameter, D, and measured pitch, P, and disc area in square feet of the propeller $= \frac{\pi D^2}{4}$, the values

$$C_A = \frac{A V^3}{\text{I.H.P.}},$$

$$C_R = \frac{RD}{V},$$

are obtained. Taking this value of C_A as the value of the ordinate at the abscissa value H.A.÷D.A. of the actual propeller, and projecting across to the ordinate erected at the abscissa whose value is that of the C_R so obtained, a point is plotted. Through this point draw a line parallel to the line A–B, Sheet 16. This line will be the locus of all values of C_A and C_R for any value of M. Where this locus crosses the value of $P \div D$ of the actual propeller will be the approximate location of the propeller on the chart.

Now, in the formula

$$C_R = \frac{R \times D}{M \times S},$$

the values C_R, R, D, and S = speed of ship are known, and from these known values the value of M can be obtained as

$$M = \frac{R \times D}{C_R \times S}.$$

Where curves of I.H.P.—Speed, Revolutions—Speed are available through a range of speeds, a corresponding curve of M can be laid down. If M continues at a nearly constant value through a long range of speeds and then suddenly rises and continues to rise rapidly as higher speeds are reached, it is an indication that the propeller is breaking down and that an improvement at the higher speeds can be expected should the acting surface of the propeller be increased.

From the results of trials of vessels similar to that for which it may be desired to design a propeller, the approximate best value of M to use may be determined. Taking the estimated I.H.P., S(speed of ship) $\times M$, and varying disc areas of propellers up to as high a diameter as may be fitted, find the value of C_A for each of these assumed disc area values and find the corresponding values of C_R for the desired revolutions and the values of D corresponding to the different values of disc area.

Assuming a value for H.A.÷D.A. of any value, say, .34, take

the values of C_A and C_R corresponding to any one diameter value and plot it on the chart and through the point thus obtained draw a line parallel to the line of constant helicoidal area ratio, C–D. The line so obtained will contain all the diameters for H.A. ÷ D.A. = .34, but C_A and C_R varying. Where this line crosses the line of maximum efficiency will be the position of the desired propeller unless the resultant diameter is too great, when a larger value of H.A. ÷ D.A. should be tried.

From the location of the obtained propeller on the chart, can be obtained:

Diameter. Deduced from value of C_R.
$P \div D$. Given on Chart.
H.A. ÷ D.A. Assumed in computation.
Efficiency of propeller but not propulsive coefficient.

The pitch will be $= \text{Diameter} \times \dfrac{P}{D}$.

The projected area of the propeller will equal

$$\frac{\text{Helicoidal Area}}{\sqrt{1 + .425\left(\frac{P}{D}\right)^2}} \text{ for elliptical blades.}$$

CHAPTER VI

THIRD METHOD OF DESIGN: DESIGN BASED ON ACTUAL TRIALS OF FULL-SIZED PROPELLERS IN SERVICE OVER CAREFULLY MEASURED COURSES. THE DYSON METHOD

In November, 1915, Sir Archibald Denny in reading a paper on "Model Tank Experiments on Naval Propellers," rather emphatically stated that *in the future the rules for the correct designing of propellers should be derived from data carefully taken from the trials of smooth bottom vessels carefully run over accurately measured deep water courses.*

This statement by such a noted authority is in line with the views of the author of this book and outlines exactly the plan which he has been following since 1901 when he first took up the study of propellers seriously. The results obtained from these years of study will now be given as clearly as it is in his power to present them.

All screw propellers when working under similar conditions of resistance arrange themselves in one great family in which the position of any particular propeller is fixed by its diameter, its pitch and its projected area ratio, the latter fixing the dimensions of the thrusts and the resultant tip speeds, and most important of all, the efficiency; the propulsive coefficient being this efficiency as modified by the existing hull conditions.

Let this condition of equal resistance be called the Basic Condition, as it applies equally to all propellers.

SHEET 20, BASIC CONDITION

On Sheet 20, are shown the curves of Indicated Thrust per square inch of disc area of the propeller, $I.T._D$; the curve of Tip Speeds in feet per minute corresponding to these values of

$I.T._D$, marked T.S.; the curves of 1 minus the apparent slip as modified by the different values of slip block coefficient from unity to the phantom ship of zero block, marked $1-S$; and finally the values of the propulsive coefficients which can be obtained at this condition of standard resistance, the hulls having the minimum losses possible due to thrust deduction; the propulsive coefficient curve is marked P.C., and the condition of Basic hull efficiency or Basic thrust deduction corresponds to the value $K=1$.

These curves are laid down on values of projected area ratio, P.A.÷D.A., as abscissas.

The Basic Curve is that of $I.T._D$, and is represented by the equation

$$I.T._D = 28.54\left(\frac{P.A.}{D.A.}\right)^{1.7},$$

where $I.T._D$ = Indicated thrust per square inch of disc area of the propeller $= (33{,}000 \times I.H.P.) \div \left(\text{Pitch} \times \text{Revolutions} \times \frac{\pi}{4} \times \overline{\text{Diameter of Propeller in feet}}^{\,2} \times 144\right)$.

T.S. = The tip-speeds of the propellers in feet per minute $= \text{Revolutions} \times \pi \times \text{Diameter in feet}$, corresponding to these values of $I.T._D$ and of P.A. ÷D.A. are also shown as a curve. It should be thoroughly borne in mind that these tip speeds and the corresponding values of $I.T._D$ are coincident only under the conditions of resistance for the Chart. Should the resistance change the tip speed may change and the corresponding value of $I.T._D$ will also change for a constant value of I.H.P., but the value of $I.T._D \times T.S.$ or $I.T._D \times \text{Revolutions}$, will remain constant.

$1-S = 1 -$ apparent slip under Basic condition of resistance $= P.T._p \div E.T._p$ = Propulsive thrust divided by effective thrust. These curves are shown for different values of slip block coefficient varying

from unity to the zero value of the phantom ship.

P.C. = Propulsive coefficient of the propeller = Basic E.H.P. ÷ Basic I.H.P. = Basic E.H.P. ÷ (Basic S.H.P. ÷ .92), the ratio between I.H.P. and S.H.P. for well-designed, well-adjusted and well-lubricated reciprocating engines without attached pumps being taken as S.H.P. = .92 I.H.P.

The curve of P.C. is seen to rise rapidly from zero value of P.A. ÷ D.A., at which point the propeller would be of infinite diameter and pitch but of zero tip speed, to a maximum value where P.A. ÷ D.A. = .2. It then falls gradually to P.A. ÷ D.A. = .25, after which its rate of fall increases until P.A. ÷ D.A. = .54, where it rapidly decreases until at P.A. ÷ D.A. = .55, the value of P.C. has reached its minimum, and this minimum value it retains up to the limit of design which is taken as P.A. ÷ D.A. = .650. For hydraulic propellers the ordinary range of design extends from approximately P.A. ÷ D.A. = .2 to P.A. ÷ D.A. = .650, the propulsive efficiency decreasing as the thrusts, tip speeds and projected area ratios increase. When air ship propellers are considered, however, they are found to lie to the left of the vertex of the propulsive efficiency curve, and the action is the direct opposite, that is as the thrusts, tip speeds and projected area ratios increase, the propulsive efficiencies increase with them.

Before going further into the subject of design it will be well to give the

Definitions of Terms and Abbreviations used in the Work

I.H.P.$_p$ = Indicated Horse-power of Propelling Engine on one propeller, without thrust deduction.

S.H.P.$_p$ = B.H.P.$_p$ = .92 I.H.P.$_p$ = Shaft or Brake Horse-power applied to the line shafting and measured by torsion of shaft abaft the thrust bearing, without thrust deduction.

e.h.p. = Net effective or tow-rope horse-power required to tow the hull at any given speed, the hull being fitted with all appendages. e.h.p. ÷ number of propellers equals the effective horse-power that must be delivered by one propeller.

I.H.P. = Indicated Horse-power which can be delivered by the propeller under Basic conditions.

S.H.P. = B.H.P. = .92 I.H.P. = Shaft or Brake Horse-power absorbed by the propeller under Basic conditions.

E.H.P. = Effective (tow-rope) Horse-power which can be delivered by the propeller under Basic conditions.

K = Thrust deduction factor due to form of hull and location of propeller when net effective thrusts do not exceed critical thrusts shown on Sheet 22. Sheet 19.

v = Actual speed of vessel corresponding to e.h.p.

V = Basic speed corresponding to E.H.P.

e.h.p. ÷ E.H.P. = Net load factor under which the propeller is operating.

$v \div V$ = Speed factor under which the propeller is operating.

Z = Value of exponent in 10^Z in equation for "power at other than Basic condition." It has the following values:

Table VI

VALUES OF Z TO BE ADDED OR SUBTRACTED FROM LOG I.H.P.

$\frac{\text{e. h. p.}}{\text{E.H.P.}}$	Z		$\frac{\text{e. h. p.}}{\text{E.H.P.}}$	Z		$\frac{\text{e. h. p.}}{\text{E.H.P.}}$	Z	
	Empir.	Calc.		Empir.	Calc.		Empir.	Calc.
.01	.....	2.0828	.1	1.0268	1.0414	1.15	.065	.0632
.015	.....	1.8994	.2	.728	.7279	1.2	.084	.0825
.02	.....	1.7693	.3	.5493	.5445	1.25	.102	.1009
.025	1.603	1.6684	.4	.4238	.4144	1.3	.1195	.1186
.03	.....	1.58594	.5	.3267	.3135	1.35	.1361	.1357
.04	.....	1.45582	.6	.2432	.2310	1.4	.1518	.1521
.05	1.359	1.3549	.7	.1690	.1613	1.45	.1676	.1681
.06	.....	1.27244	.8	.1065	.1009	1.5	.1823	.1834
.07	.....	1.20271	.9	.05402	.0477	1.55	.197	.1932
.075	1.165	1.1715	1.0	0	0	1.6	.21	.2126
.08	.....	1.1423	1.05	.0225	.0221	1.65	.225	.2265
.09	.....	1.0891	1.1	.0450	.0431	1.7	.239	.2400
.....	.....	.....	......	.....	.....	1.75	.250	.2531

e.h.p. = Gross effective horse-power which would be delivered by the propeller with a total power I.H.P._d if no thrust deduction existed.

e.t. = Net effective thrust.

e.t. = Gross effective thrust.

E.T. = Critical effective thrust.

D = Diameter of propeller in feet.

P = Pitch of propeller in feet.

T.S. = Tip-speed of Propeller in feet per minute under Basic conditions.

P.A. ÷ D.A. = Projected Area ratio of 3-bladed propeller.

$\frac{4}{3}$(P.A. ÷ D.A.) = Projected Area ratio of 4-bladed propeller.

$\frac{2}{3}$(P.A. ÷ D.A.) = Projected Area ratio of 2-bladed propeller.

P.C. = Basic propulsive coefficient for *total* projected area ratio no matter what the number of blades of the propeller.

p.c. = Actual propulsive coefficient delivered by the propeller.

The value of p.c. depends upon the value of P.C., of K, and of the load factor e.h.p. ÷ E.H.P., and of $v \div V$. Where $v \div V$ is not less than the values corresponding to the curve marked "Critical Thrusts" on Sheet 22, the value of p.c. depends upon the first three factors, only.

The relative values of p.c. and P.C. for varying values of e.h.p. ÷ E.H.P., where the values of $v \div V$ are equal to or greater than those corresponding to the critical thrust, *E.T.*, disregarding the value of K, are given in the following table; the actual values of propulsive coefficient would be reduced, however, inversely as the value of K:

TABLE VII

VALUES OF p.c. ÷ P.C. FOR VARYING VALUES OF e.h.p. ÷ E.H.P.

e. h. p. / E.H.P.	p. c. / P.C.	e. h. p. / E.H.P.	p. c. / P.C.	e. h. p. / E.H.P.	p. c. / P.C.	e. h. p. / E.H.P.	p. c. / P.C.	e. h. p. / E.H.P.	p. c. / P.C.
.025	1.002	.4	1.061	1.0	1.0	1.3	.9873	1.6	.9843
.05	1.143	.5	1.061	1.05	.997	1.35	.9868	1.65	.9838
.075	1.097	.6	1.05	1.10	.9917	1.4	.9863	1.7	.9833
.1	1.064	.7	1.033	1.15	.9901	1.45	.9858	1.75	.9828
.2	1.069	.8	1.022	1.2	.989	1.5	.9853	1.8	.9823
.3	1.063	.9	1.019	1.25	.9884	1.55	.9848	1.85	.9818

These values of p.c. ÷ P.C. are obtained from the values of Z and of e.h.p. ÷ E.H.P. by means of the following equation:

Log p.c. = log P.C. + log $\left(\frac{\text{e.h.p.}}{\text{E.H.P.}}\right) + Z$ where Z is additive for values of e.h.p. ÷ E.H.P., less than unity, and subtractive for those greater than unity. The empirical values of Z have been used but the calculated are better.

Above the critical thrusts (Sheet 22), these values of p.c. ÷ P.C. and the corresponding values of Z only hold up to a point where the value $v \div V$ is slightly less than e.h.p. ÷ E.H.P., this limiting point being taken as

$$v \div V = \text{e.h.p.} \div (1.15 \text{ E.H.P.}),$$

or, in other words, an increase of 15 per cent in the effective thrust over the effective thrusts for the basic condition of the propeller. After passing this point the value of Z changes very rapidly, due to cavitation of the suction column, causing a rapid increase in power and a corresponding decrease in the value of the propulsive coefficient. Where thrust deduction exists, the final ratio of actual to basic propulsive coefficient for any projected area ratio of propeller becomes at thrusts equal to or less than those corresponding to the Curve Critical Thrusts, *E.T.*, (Sheet 22), and at positions above that curve,

$$\text{p.c.} \div \text{K.P.C.}$$

where the effective thrusts are greater than the critical thrusts, *E.T.*, the final value of p.c. becomes log p.c. = log P.C. + log $\left(\frac{\text{e.h.p.}}{\text{E.H.P.}}\right) - \log \text{K} - \log\left(\frac{\text{e.t.}}{E.T.}\right)^x + Z.$

s = Apparent slip of propeller at speed v.

S = Apparent slip of propeller at basic speed V under basic conditions of power and resistance.

I.T. = Total indicated thrust exerted by the propeller under basic conditions,

$$= \frac{\text{I.H.P.} \times 33{,}000}{P \times R} = \frac{\text{S.H.P.} \times 33{,}000}{.92 \times P \times R}.$$

$R = \text{T.S.} \div \pi D =$ Basic revolutions.

$\text{I.T.}_{D} = \text{I.T.} \div \left(144 \times \frac{\pi}{4} D^2\right) =$ Indicated thrust per square inch of disc area under basic conditions.

P.T. = Total propulsive thrust of the propeller under basic conditions,

$$= \frac{\text{E.H.P.} \times 33{,}000}{P \times R} = \text{I.T.} \times \text{P.C.}$$

There is often a different expression for indicated thrust given than the one above, so in order to avoid confusion, the writer has adopted the term "Speed Thrust," to differentiate between the two. V.T. = Speed thrust under basic conditions,

$$= \frac{\text{I.H.P.} \times 33{,}000}{V \times 101.33} = \frac{\text{S.H.P.} \times 33{,}000}{.92 \times V \times 101.33}.$$

E.T. = Effective thrust under basic conditions,

$$= \frac{\text{E.H.P.} \times 33{,}000}{V \times 101.33}.$$

The ratio between the indicated and speed thrusts and between the propulsive and effective thrusts is

$$1 - S = \frac{\text{I.T.}}{\text{V.T.}} = \frac{\text{P.T.}}{\text{E.T.}} = \frac{V \times 101.33}{P \times R}.$$

Derivation of Formulas under Basic Conditions

Formula for Pitch:

$$P \times R = P \times \frac{\text{T.S.}}{\pi D}.$$

Speed of ship in feet per minute $= 101.33 \times V$

$$= (1 - S) P \times R = \frac{P \times \text{T.S.} \times (1 - S)}{\pi D}.$$

Apparent slip of propeller in feet per minute under Basic conditions

$$=(P\times R)-(P\times R)(1-S)=P\times R\times S=\frac{P\times \text{T.S.}\times S}{\pi D}.$$

$$\therefore\ P=\frac{P\times R\times \pi D}{\text{T.S.}}=\frac{101.33\times V\times \pi\times D}{\text{T.S.}\times(1-S)}.$$

Derivation of Equation for D:

$$\text{I.T.}=(\text{I.H.P.}\times 33{,}000)\div(P\times R).$$

$$\text{I.T.}_{\cdot D}=\text{I.T.}\div\left(144\times\frac{\pi}{4}D^2\right)=\frac{\text{I.H.P.}\times 33{,}000}{36\times\pi\times D^2\times P\times R}.$$

$$\therefore\ D^2=\frac{\text{I.H.P.}\times 33{,}000}{36\pi\times \text{I.T.}_{\cdot D}\times P\times R}=\frac{291.8\times \text{I.H.P}}{\text{I.T.}_{\cdot D}\times P\times R}.$$

$$\therefore\ D=\sqrt{\frac{291.8\times \text{I.H.P.}}{\text{I.T.}_{\cdot D}\times P\times R}}=\sqrt{\frac{291.8\times \text{I.H.P.}\times(1-S)}{\text{I.T.}_{\cdot D}\times V\times(101.33)}}$$

$$=\sqrt{\frac{2.88\times \text{I.H.P.}\times(1-S)}{\text{I.T.}_{\cdot D}\times V}}.$$

This formula applies to three-bladed propellers only, and requires modifications, as follows, for four- and for two-bladed ones:

Four-bladed

$$D=\sqrt{\frac{252.41\times \text{I.H.P}}{\text{I.T.}_{\cdot D}\times P\times R}}=\sqrt{\frac{252.41\times \text{I.H.P.}\times(1-S)}{\text{I.T.}_{\cdot D}\times V\times 101.33}}$$

$$=\sqrt{\frac{2.491\times \text{I.H.P.}\times(1-S)}{\text{I.T.}_{\cdot D}\times V}}.$$

Two-bladed

$$D=\sqrt{\frac{389\times \text{I.H.P.}}{\text{I.T.}_{\cdot D}\times P\times R}}=\sqrt{\frac{389\times \text{I.H.P.}\times(1-S)}{\text{I.T.}_{\cdot D}\times V.\times 101.33}}$$

$$=\sqrt{\frac{3.84\times \text{I.H.P.}\times(1-S)}{\text{I.T.}_{\cdot D}\times V}}.$$

This may be expressed by the following general equation:

$$D=\sqrt{\frac{B\times \text{I.H.P.}}{\text{I.T.}_{\cdot D}\times P\times R}}=\sqrt{\frac{\text{B}\times \text{I.H.P.}\times(1-S)}{\text{I.T.}_{\cdot D}\times V}},$$

also

$$P \times D = \frac{B \times \pi \times \text{I.H.P.}}{\text{I.T.}_{D} \times \text{T.S.}},$$

whence for any value of D,

$$P = \frac{B \times \pi \times \text{I.H.P.}}{D \times \text{I.T.}_{D} \times \text{T.S.}}$$

and

$$V = \frac{P \times \text{T.S.} \times (1 - S)}{\pi \times D \times 101.33}.$$

Method of Changing from Basic Conditions to Other Conditions of Resistance

It has been explained how for Basic conditions of resistance of any vessel, and without thrust deduction, and with a given propeller, the vessel will run at a speed V with an indicated horse-power, I.H.P., shaft horse-power S.H.P., with which power an effective (tow-rope) horse-power E.H.P., will be delivered. The revolutions under these conditions will be R and the tip-speed T.S.

Should the speed be reduced by reducing the power of the engines, by increasing the displacement, by fouling of bottom, by condition of wind or sea, etc., or should the opposite exist and the speed be increased, the conditions of resistance will differ from the Basic conditions and the following changes will occur in the propeller performance from those which existed under the Basic condition:

1. *Basic condition of resistance for* V, but

I.H.P.$_{p}$ = Engine power	Reduced	Increased
e.h.p. = Effective H.P. delivered	Reduced	Increased
R_D = Revolutions	Reduced	Increased
t.s. = Tip-speed	Reduced	Increased
v = Speed	Reduced	Increased

Where thrust deduction exists, the new engine power will be I.H.P.$_{d} = K \times$ I.H.P.$_{p}$.

The effect on the thrusts will be variable, depending upon the values of the speed and load factors, $v \div V$ and e.h.p. $\div$ E.H.P.

2. *Basic condition of Power constant* but

v = Speed for E.H.P.	Reduced	Increased
I.H.P. = Power	Constant	Constant
E.H.P. = Effective Horse-power	Constant	Constant
R_a = Revolutions for E.H.P.	Reduced	Increased
t.s. = Tip-speed	Reduced	Increased
Thrusts	Increased	Reduced

Where K is greater than unity, I.H.P. becomes $K \times$ I.H.P.

In these changes of condition, so long as the Basic I.H.P. remains constant the corresponding E.H.P. also remains constant no matter what the speeds of ship, provided, however, that *the speed of ship is not so low as to produce serious augmentation of thrusts.* When such thrusts are attained the apparent slips will increase rapidly, while delivering the same E.H.P., and this increase of slip will be accompanied by a decrease in propulsive efficiency.

The above paragraph is justified by the comparison of very accurate trial results of several vessels which were of sufficiently fine after body and where the propellers were so well located as to practically insure a value of unity for K. In these cases the agreement between the actual indicated, shaft and effective (tow-rope) horse-powers and those of the basic conditions of the propellers were so close as to lead to the following conclusion:

LAW OF EFFICIENCY. *Should a screw propeller working in the wake of a vessel deliver a certain effective (tow-rope) horse-power with a certain indicated or shaft horse-power under any given condition of resistance, it will deliver the same effective with the same indicated or shaft horse-power under any other condition of resistance so long as it is operating in the wake of the same hull, and so long as the effective thrusts are well below the "Critical effective thrusts."*

The *law of propulsive efficiency* just given, renders it possible, where a vessel has been tried up to and beyond the speed for which the effective (tow-rope) horse-power is equal to the Basic E.H.P. of the propeller used, to obtain the value of K at once, as,—

The actual indicated horse-power or shaft horse-power required to deliver E.H.P. $= K \times$ I.H.P. or $K \times$ S.H.P. (Basic powers), from which at once results

$$K = \frac{\text{Actual indicated or shaft horse-power}}{\text{Basic indicated or shaft horse-power}}.$$

Sheet 21. *The Power Corrective Factor Z.*

In arriving at a satisfactory series of values of corrective factors to use in estimating changes in power due to changes in conditions of load from the Basic condition, many different forms of equations were tried, using the measured mile trial data of very long and fine vessels, tried in deep water, and where the trials were conducted in such a manner as to give confidence in the trial data tabulated.

All the trials that have been used had at least three runs for each point of the speed-revolution and speed-power curves while the highest point plotted for each vessel was obtained as a mean of five runs. In obtaining the mean of each set of runs, the following method of averaging was used:

For a five-run point:

Run No. 1	North	$1 \times$ Power	$1 \times$ Revolutions
Run No. 2	South	$2 \times$ Power	$2 \times$ Revolutions
Run No. 3	North	$2 \times$ Power	$2 \times$ Revolutions
Run No. 4	South	$2 \times$ Power	$2 \times$ Revolutions
Run No. 5	North	$1 \times$ Power	$1 \times$ Revolutions
Mean........	....	$\frac{\Sigma \text{ Power}}{8}$	$\frac{\Sigma \text{ Revolutions}}{8}$

For a three-run point:

Run No. 1	North	$1 \times$ Power	$1 \times$ Revolutions
Run No. 2	South	$2 \times$ Power	$2 \times$ Revolutions
Run No. 3	North	$1 \times$ Power	$1 \times$ Revolutions
Mean.........	....	$\frac{\Sigma \text{ Power}}{4}$	$\frac{\Sigma \text{ Revolutions}}{4}$

The form of equation finally obtained was of the form

$$\text{I.H.P.}_v = \text{I.H.P.} \times \left(\frac{v}{V}\right)^w,$$

where v is any speed of vessel and I.H.P.$_v$ the indicated horse-power for this speed. When the thrust deduction factor K exceeds unity, the actual indicated horse-power for v becomes $K \times$I.H.P.$_v$ = I.H.P.$_a$.

Designating the effective (tow-rope) horse-power necessary to obtain a speed v, by e.h.p., and taking values of v for several trial vessels for the load ratios e.h.p. ÷ E.H.P. = .025, .05, .1, .2, .3, .4, etc., up to as high a ratio as the available data would give, and solving the equation

$$w = (\log \text{I.H.P.} - \log \text{I.H.P.}_v) \div (\log. V - \log v),$$

I.H.P.$_v$ being the actual indicated horse-power for v in cases where $K = 1$ and being equal to that horse-power divided by K when K was greater than unity, a series of curves were obtained for the different load ratios given.

Taking $Z = w(\log V - \log v)$, it was found that for each of these curves of w, Z had practically a constant value which depended upon the value of e.h.p. ÷ E.H.P. These values are given in the preceding table of Z and are also shown as a curve on Sheet 21.

The final equation for indicated or shaft horse-power for any other than Basic conditions becomes

$$\left.\begin{matrix}\text{I.H.P.}_v \\ \text{S.H.P.}_v\end{matrix}\right\} = \left\{\begin{matrix}\text{I.H.P.} \\ \text{S.H.P.}\end{matrix}\right\} \div 10^Z,$$

which expressed in logarithms becomes

$$\log \left\{\begin{matrix}\text{I.H.P.}_v \\ \text{S.H.P.}_v\end{matrix}\right\} = \log \left\{\begin{matrix}\text{I.H.P.} \\ \text{S.H.P.}\end{matrix}\right\} \pm Z,$$

Z being subtractive when e.h.p. ÷ E.H.P. is less than unity and additive when greater.

The table of Z values gives two columns of values, one being the empirical values obtained from actual trials of vessels and

propellers, while the second are the calculated values given by the dotted curve of Sheet 21.

The equation fitting this curve is

$$10^Z = \frac{C}{\left(10\frac{\text{e.h.p.}}{\text{E.H.P.}}\right)^{\log C}},$$

and the value of C is approximately 11, so that putting the equation in the logarithmic form,

$$Z = 1.0414 - 1.0414\left(\log\frac{10 \text{ e.h.p.}}{\text{E.H.P.}}\right).$$

This equation is the fundamental equation for the estimate of power and can be used no matter what value of effective power is used as a base, so long as the effective thrusts do not exceed the "Critical Thrusts, $E.T.$" (Sheet 22). Thus, if there is available for use the actual indicated or shaft horse-power, $\text{I.H.P.}_a = K\ \text{I.H.P.}_p$ or $\text{S.H.P.}_a = K\ \text{S.H.P.}_p$ for any speed v for which the tow-rope (effective) power is e.h.p., the value of the indicated or shaft horse-power necessary for any other speed v_1 requiring an effective horse-power e.h.p._1 where the vessel is in same condition of hull as to displacement and condition of bottom and where weather conditions are similar, may be computed and the basic characteristics of the propeller used can be entirely disregarded.

This is shown by the table on p. 68, where the basic conditions of design and variation of load are given and then the changes that occur when the actual basic load is assumed as .5 of the basic design load.

The final logs of the estimated horse-power factor in Column 9 are seen to be the same as those in Column 4.

The power corrective factors Z as given, however, only hold for certain conditions of e.t. and of e.h.p. ÷ E.H.P., and these conditions are shown by the curve marked "Critical Thrusts," and above on Sheet 22. This curve is erected on values of e.h.p ÷ E.H.P. as abscissas. Should the actual value of e.t. be less than the critical value $E.T.$ corresponding to the value of e.h.p. ÷ E.H.P.,

there may be a slight increase in the value of the propulsive coefficient. Should it be greater, however, the p.c. will be gradually decreased, the decrease becoming more rapid as the value of e.t. increases (Sheet 22). Under these conditions the power equation becomes I.H.P.$_p$ = I.H.P. $\div 10^Z$.

$$\text{I.H.P.}_d = \text{I.H.P.}_p \times K \times \left(\frac{\text{e.t.}}{E.T.}\right)^z = \text{I.H.P.}_p \times K \times M$$

TABLE VIII

1	2	3	4	5		6	7	8	9
Log of Basic I.H.P.	$\frac{\text{e.h.p.}}{\text{E.H.P.}}$	Z	Log of I.H.P.$_p$ factor $1-Z$	Log of I.H.P.$_p$ factor for .5 Load $1-Z=1$		$\frac{\text{e.h.p.}}{.5\text{E.H.P.}}$	Z^1 for Col. 6	$1-Z^{11}=$ $1-Z-Z^1=$ Log of I.H.P.$_d$	Log of I.H.P.$_p$ factor $1=Z^1$
1	.025	1.6684	8.3316	9.6865	1	.05	1.3549	9.6865−1.3549	8.3316
1	.05	1.3549	8.6451	9.6865	1	.1	1.0414	9.6865−1.0414	8.6451
1	.1	1.0414	8.9586	9.6865	1	.2	.7279	9.6865− .7279	8.9586
1	.2	.7279	9.2721	9.6865	1	.4	.4144	9.6865− .4144	9.2721
1	.3	.5445	9.4555	9.6865	1	.6	.231	9.6865− .231	9.4555
1	.4	.4144	9.5856	9.6865	1	.8	.1009	9.6865− .1009	9.5856
1	.5	.3135	9.6865	9.6865	1	1.0	0	9.6865− 0	9.6865
1	.6	.231	9.769	9.6865	1	1.2	.0825	9.6865− .0825	9.7690
1	.7	.1613	9.8387	9.6865	1	1.4	.1521	9.6865− .1521	9.8386
1	.8	.1009	9.8991	9.6865	1	1.6	.2126	9.6865− .2126	9.8991

ESTIMATE OF REVOLUTIONS FOR OTHER THAN BASIC CONDITIONS OF RESISTANCE

For making the estimate of the revolutions due to the change in conditions resulting from change in power accompanied by corresponding change in speed, the following equations derived by Commander S. M. Robinson, U. S. Navy, and which will here be denoted as the "Robinson Equations for Revolutions" are used. The forms for estimating are

$$s = S\frac{\text{I.H.P.}_d \times V^n}{\text{I.H.P.} \times v^{n}} = S \times \frac{K \times \text{I.H.P.}_p \times V^n}{\text{I.H.P.} \times v^{n}} = S \times \frac{K \times V^n}{10^Z \times v^{n}},$$

where all of the terms except y have the same meaning as given in the list of terms.

The values of $y \times \log$ speed, denoted by $\log A$, are shown on Sheet 21.

The logarithmic form is:

log of apparent slip at speed v = log of apparent slip under Basic condition + log of actual indicated or shaft horse-power + $y_1 \times$ log of Basic Speed − log of Basic indicated or shaft horse-power − $y_2 \times$ log of actual speed.

This in its final form becomes

$$\log s = \log S + \log K + \log A_V - \log A_v \mp Z,$$

Z being subtractive for values of e.h.p. ÷ E.H.P. less than unity and additive for values greater than unity.

Having the apparent slip for speed v, the equation for revolutions is

$$R_d = \frac{v \times 101.33}{\text{Pitch} \times (1-s)} = \frac{v \times 101.33}{P \times (1-s)}.$$

The values of y in the Robinson equation are given by a curve expressed by the following equation:

$$y = \frac{v}{v-1}\left[2.626 + \frac{.009129}{.0000015575(v-25)^4 + .04368}\right]$$

$$= \frac{v}{v-1}\left[2.626 + \frac{5861.3}{28045 + (v-25)^4}\right].$$

The equation for apparent slip s_1 at a speed v_1 and indicated horse-power I.H.P.$_{d_1}$, in terms of the Basic conditions being,

$$s_1 = S \times \frac{\text{I.H.P.}_{d_1} \times V^y}{\text{I.H.P.} \times v_1^{y_1}}.$$

the apparent slip s_2, for a speed v_2, and indicated horse-power I.H.P.$_{d_2}$, will be

$$s_2 = S \times \frac{\text{I.H.P.}_{d_2} \times V^y}{\text{I.H.P.} \times v_2^{y_2}},$$

or

$$s_2 = s_1 \frac{\text{I.H.P.}_{d_2} \times v_1^{y_1}}{\text{I.H.P.}_{d_1} \times v_2^{y_2}},$$

and this is the *fundamental* equation for apparent slip.

The above equation for apparent slip in its final form only holds, however, up to certain coincident values of e.h.p. ÷ E.H.P. and e.t., that is up to fixed values of net e.t., and these values are shown by the curve on Sheet 22, marked *E.T.* For any higher value of e.t. than given by this curve for any given value of e.h.p. ÷ E.H.P., the final equation for apparent slip becomes

$$s=S\times\frac{\text{I.H.P.}_d\times V^v}{\text{I.H.P.}\times v^y}=S\times\frac{K\times A_V}{10^Z\times A_v}\times\left(\frac{\text{e.t.}}{E.T.}\right)^x.$$

Thus, suppose $v\div V=.5$, e.h.p. ÷ E.H.P. = .4. The value of e.t. ÷ E.T. for these values of $v\div V$ and e.h.p. ÷ E.H.P. is .8.

The Critical value *E.T.* of e.t. ÷ E.T. for e.h.p. ÷ E.H.P. = .4, is .627, therefore,

$$s=S\times\frac{\text{I.H.P.}_d\times V^v}{\text{I.H.P.}\times v^y}=S\frac{K\times A_V}{10^{.4144}\times A_v}\times\left(\frac{\text{e.t.}}{E.T.}\right)^x,$$

and the value $\left(\frac{\text{e.t.}}{E.T.}\right)^x$ for $\frac{\text{e.h.p.}}{\text{E.H.P.}}=.4$ and $\frac{v}{V}=.5$, is seen from Sheet 22, to be equal to $\left(\frac{1}{.57}\right)$.

Relation between Power and Revolutions when the vessel is prevented from advancing.

When a vessel is secured to a dock so that after the securing hawsers are taut there can be no further motion of the vessel through the water, the conditions of operation of the propeller become radically different from those existing when the vessel is in free route.

Different as are the conditions there still remains a definite relation between the actual conditions and the chart (Basic) condition of the propeller so far as power and revolutions are concerned, and this relation is expressed by the following logarithmic equation

$$\therefore\quad \text{Log}\left(\frac{R}{R_d}\right)=.2794\log\left(\frac{\text{I.H.P.}}{\text{I.H.P.}_d}\right)+.14246=.2794Z+.14246,$$

or

$$\frac{R}{R_d}=1.3882\left(\frac{\text{I.H.P.}}{\text{I.H.P.}_d}\right)^{.2794}$$

but

$$\frac{\text{I.H.P.}}{\text{I.H.P.}_{d}} = 10^{Z}.$$

$$\therefore \frac{R}{R_d} = 1.3882 \times 10^{.2794Z}$$

or

$$R_d = \frac{R}{1.3882 \times 10^{.2794Z}},$$

where R_d and I.H.P.$_d$ are the revolutions and power for the actual conditions and R and I.H.P. those for the Basic condition of the propeller.

When I.H.P. = I.H.P.$_d$,

$R_d = R \div 1.3882$, and this reduction in revolutions for a power equal to the Basic power of the propeller is due entirely to the elimination of the effect of the Basic speed V.

Should we have two conditions of revolutions and power, R_{d_1}, I.H.P.$_{d_1}$ and R_{d_2} and I.H.P.$_{d_2}$, the vessel being secured to a dock for both conditions, the relation between revolutions and power will be expressed by

$$\log\left(\frac{R_{d_1}}{R_{d_2}}\right) = .2794\left(\log \frac{\text{I.H.P.}_{d_1}}{\text{I.H.P.}_{d_2}}\right) = .2794(Z_2 - Z_1),$$

or

$$R_{d_1} = R_{d_2} \times \left(\frac{\text{I.H.P.}_{d_1}}{\text{I.H.P.}_{d_2}}\right)^{.2794} = R_{d_2} \times 10^{.2794(Z_2 - Z_1)}.$$

To find apparent slip or approximate power, power and speed or apparent slip and speed known.

From the fundamental equation it will be seen at once how, having the apparent slip, speed of ship and horse-power, the apparent slip or the horse-power for any other speed may be obtained, provided the conditions of hull, displacement and weather are the same, thus:

If the power and speed are known, the apparent slip can be found by the fundamental equation as already given.

To find the approximate power, the speed and apparent slip being known:

$$\text{I.H.P.}_{d_2}=\frac{s_2\times \text{I.H.P.}_{d_1}\times v_2^{v_2}}{s_1\times v_1^{v_1}}=\frac{s_2\times \text{I.H.P.}\times v_2^{v_2}}{S\times V^{v}}.$$

This neglects the effect of variation of thrusts over critical thrusts.

Should $\text{I.H.P.}_{d_2}=\text{I.H.P.}_{d_1}$,

$$s_2=s_1\times\frac{v_1^{v_1}}{v_2^{v_2}},$$

while, should $v_2=v_1$ but I.H.P._{d_2} be greater or less than I.H.P._{d_1}

$$s_2=s_1\times\frac{\text{I.H.P.}_{d_2}}{\text{I.H.P.}_{1}},$$

that is, the approximate apparent slips for constant speed but varying power will vary almost directly as the power, or in other words, *where the speed of a vessel is constant but the power required for that speed is variable, the revolutions required will vary directly as the power unless the critical thrusts are exceeded.*

Effect of Variations in Mechanical Efficiency of Engine

Sheet 20, as constructed is based on a mechanical efficiency of .92 for reciprocating engines, and the Basic values of the propulsive coefficients as given on this sheet, only fit this mechanical efficiency. The relations between I.H.P., S.H.P., and E.H.P. being expressed by

$$\text{I.H.P.}=\frac{\text{S.H.P.}}{.92}=\frac{\text{E.H.P.}}{\text{P.C.}},$$

should the mechanical efficiency differ from .92, the relations between these powers must be corrected accordingly; thus, suppose a mechanical efficiency of only .85 is expected, then

$$\text{I.H.P}=\frac{\text{S.H.P.}}{.85}=\frac{\text{E.H.P.}\times .92}{.85\times \text{P.C.}},$$

and the I.H.P. to use in the equation for diameter would be only $\frac{85}{92}\times$the actual I.H.P. of the main engines where the propeller is being designed for Basic conditions of resistance.

Number of Blades and their Effect on Efficiency

The Design Sheet 20 has been developed from the data of performances of three-bladed propellers, and, therefore, a correction must be applied in the calculations for diameter and for estimated propulsive coefficients if it should be desired to use any other number of blades.

Should a four-bladed propeller be desired, the total indicated horse-power required for any given number of revolutions will be the indicated horse-power required by a three-bladed propeller of the same pitch and diameter as the four-bladed one but having only three-fourths of its projected area, divided by .865, that is

$$\text{I.H.P.}_4 = \text{I.H.P.}_3 \div .865,$$

while for a two-bladed wheel the proportion becomes

$$\text{I.H.P.}_2 = \text{I.H.P.}_3 \times .75.$$

The projected area ratio of the four-bladed propeller will be equal to four-thirds of that of the three-bladed one while that of the two-bladed one will be only two-thirds of that of the three.

Thus the equations for diameter for two-, three-, and four-bladed propellers assume the following forms:

Two-bladed:

$$D=\sqrt{\frac{291.8\times \text{I.H.P.}_3}{\text{I.T.}_D\times P\times R}}=\sqrt{\frac{291.8\times \text{I.H.P.}_2}{.75\ \text{I.T.}_D\times P\times R}}=\sqrt{\frac{389\times \text{I.H.P.}_2}{\text{I.T.}_D\times P\times R}}$$

$$=\sqrt{\frac{3.84\times \text{I.H.P.}_2\times(1-S)}{\text{I.T.}_D\times V}};$$

Three-bladed:

$$D=\sqrt{\frac{291.8\times \text{I.H.P.}_3}{\text{I.T.}_D\times P\times R}}=\sqrt{\frac{2.88\times \text{I.H.P.}_3\times(1-S)}{\text{I.T.}_D\times V}}.$$

Four-bladed:

$$D=\sqrt{\frac{291.8\times \text{I.H.P.}_3}{\text{I.T.}_D\times P\times R}}=\sqrt{\frac{291.8\times \text{I.H.P.}_4\times .865}{\text{I.T.}_D\times P\times R}}$$

$$=\sqrt{\frac{252.41\times \text{I.H.P.}_4}{\text{I.T.}_D\times P\times R}}=\sqrt{\frac{2.491\times \text{I.H.P.}_4\times(1-S)}{\text{I.T.}_D\times V}}.$$

Now the Basic conditions of all three of the above propellers are, in everything but propulsive coefficient, the same as those for the three-bladed propeller having only three-fourths the projected area of the four, and one and one-half times the area of the two-bladed propellers, so that in using the design sheet the values of I.T._D, $1-S$, and T.S. for the projected area ratio of the three-bladed Basic propeller are taken.

In taking off the propulsive coefficient from the Sheet, however, it must be taken off for the actual projected area of the propeller whether it be two-, three- or four-bladed.

The usually accepted idea as to the relative propulsive efficiencies of two-, three-, and four-bladed propellers is that they stand in rank in the order given above, the two-bladed propeller being the most efficient. This is most certainly the case where the projected area ratio of the four-bladed propeller exceeds that of the three-bladed propeller, and that of the three-bladed exceeds that of the two-bladed, all being designed to deliver the same effective horse-power at the same number of revolutions, unless the projected area ratio of the propeller having the smaller number of blades should become considerably less than two-tenths. In such a case the propeller of four blades might become more efficient than that of three, and that of the three blades than that of two. In the above statement it is considered that the net values of e.t. are all below the "Critical Thrusts."

On account of the lesser number of blades, that propeller having the fewer number of blades should, generally speaking, have less loss due to eddying around the blades than would occur with an increased number, but *for constant condition of pitch and diameter, the propulsive efficiency of propellers varies with their projected area ratio, decreasing as the projected area ratio increases,*

so long as this exceeds two-tenths of the disc area, no matter what the number of blades of the propeller.

Résumé

An examination of sheets 17, 18, 19, 20, 21, 22, 22B, 23, 24, 25, and of the forms for computation of problems which will be given in succeeding chapters, will show at once that they tie together, in a consistent manner, all the elements necessary to be taken into account in the design of a propeller, thus:

From Sheet 17 is obtained the estimate of the form of the ship and the influence on wake of variation of location of the propeller in relation to the hull, to act as a guide in selecting the value of apparent slip to be used in the calculations.

From Sheet 18 is obtained an approximate estimate of the resistance of the hull appendages to apply to the estimated bare hull resistance.

From Sheet 19 is obtained the estimate of thrust deduction for the type of hull and location of propeller.

From Sheet 20 is obtained the basic factors to use in the design, that is factors of indicated thrust, tip-speed, 1 − apparent slip, and propulsive coefficient.

From Sheet 21 are obtained factors for the estimation of powers and revolutions for other than Basic conditions of the propeller, while from Sheet 22 can be ascertained the position of the propeller as regards cavitation; the correction for cavitation; and the correction of revolutions and effective horse-power for variation of speed with constant power on the propeller, and the limiting values of e.t. ÷ E.T. for safe design.

From Sheet 22B can be obtained the approximate maximum and minimum values of e.h.p. ÷ E.H.P. which should be used in calculating propellers for vessels of any slip block coefficient and desired speed, as obtained from actual results.

From Sheet 23 are obtained values of $\text{I.T.}_D \div (1-S)$ and from Sheet 24, values of $(\text{P.A.} \div \text{D.A.}) \times \text{E.T.}_p$, both for different

values of P.A.÷D.A. of the basic three-bladed propellers, these factors entering in the following equations

$$\text{I.T.}_D \div (1-S) = \frac{C \times \text{I.H.P}}{D^2 \times V}.,$$

and

$$(\text{P.A.} \div \text{D.A.}) \times \text{E.T.}_p = \frac{C \times \text{E.H.P.}}{D^2 \times V}.$$

From Sheet 25 are obtained the standard forms of blade projections which maintain the necessary constant distribution of projected surface, and also the ratio between the values of projected area ratios and the corresponding ratios of developed areas.

Problems to be Encountered in the Propeller Field

These may be classified under two general heads:

1. Problems in Analysis.
2. Problems in Design.

These classes are the converse of each other as should be expected, for according to the old saying, "It is a poor rule that will not work both ways," so if by a set of data it is possible to design a propeller to fit any condition, and the data is basically correct, then this same propeller when attacked from the other end of the problem, should return as a result, the original data. This is what actually occurs by the use of the Basic design, Sheet 20, and which sheet thus verifies its correctness.

CHAPTER VII

ANALYSIS OF PROPELLERS

By the term "analysis of a propeller" is meant an intelligent criticism of the form of its blades, of its blade sections, hub contour and an estimate of the performances of the propeller under varying conditions of load.

That part of the analysis relating to hub, blade form and section will be left until a later chapter, but that relating to the estimates of performances will be considered immediately.

Being given the form of hull, location of propeller in relation to the hull, all data concerning the characteristics of the propeller, and either the curve of tow-rope horse-power for the hull or the estimated, indicated or shaft horse-power required for any desired speed, the first step in the analysis is to obtain the Basic condition of the propeller. The method of doing this will now be explained:

Hull data: Slip block coefficient = .8.

Single screw and ship of deep draught, therefore thrust deduction factor $K = 1.26$. (Sheet 19).

Designed speed = 11 knots and 8 knots when towing.

Effective (tow-rope) horse-power for these speeds = 1000.

There are three propellers proposed from which a choice is to be made by analysis. It is desired to estimate the indicated horse-power and revolutions necessary with each propeller for the designed speeds.

The propellers are two-bladed, three-bladed and four-bladed, and are of same diameter and pitch, but the projected areas and, therefore, the projected area ratios, vary directly as the number of blades.

Derivation of Basic Condition

Number of blades	2	.3	4
P.A.÷D.A.	.2	.3	.4
Diameter	16′	16′	16′
Pitch	14′	14′	14′
Tip Speed (Sheet 20, for .3)	6650	6650	6650
$R=\text{T.S.}\div\pi D$	132.3	132.3	132.3
$P\times R$	1852	1852	1852
Slip B.C.	.8	.8	.8
$1-S$ (Sheet 20, for .3)	.941	.941	.941
$V=\dfrac{(P\times R)\times(1-S)}{101.33}$	17.2	17.2	17.2
I.T._D (Sheet 20, for .3)	3.74	3.74	3.74
$\text{I.H.P.}=\dfrac{D^2\times\text{I.T.}_D\times P\times R}{389}$	4559		
$\text{I.H.P.}=\dfrac{D^2\times\text{I.T.}_D\times P\times R}{291.8}$	.....	6077	
$\text{I.H.P.}=\dfrac{D^2\times\text{I.T.}_D\times P\times R}{252.41}$	.....	.....	7026
P.C for actual P.A.÷D.A.	.709	.682	.619
E.H.P.=I.H.P.×P.C.	3233	4145	4349
e.h.p.	1000	1000	1000
e.h.p.÷E.H.P.	.3094	.2413	.23
Z for e.h.p.÷E.H.P. (Sheet 21)	.534	.636	.658
$\text{I.H.P.}_p=\text{I.H.P.}\div 10^Z$	1333	1405	1544
$\text{I.H.P.}_d=K\times\text{I.H.P.}_p=$ Total Est. power for v	1680	1770	1946
v	11	11	11
Log A_V for $V=17.2$ (Sheet 21) = log (V^y), Curve x	3.69	3.69	3.69
Log A_v for $v=11$ (Sheet 21) = log (v^y), Curve x	3.12	3.12	3.12
$s=S\times\dfrac{K\times V^y}{10\ v^y}$	06981	.0552	.05247
$R_d=\dfrac{v\times 101.33}{P\times(1-s)}=$ Est. Rev. for v =	85.59	84.27	84.03
$v\div V$	.6395	.6395	.6395
and all propellers plot on Sheet 22, as below the "Critical Thrusts."			
Second speed v_2	8	8	8
$v_2\div V$	.465	.465	465
e.h.p.$_2$	1000	1000	1000
($E.T.$÷e.t.) (Sheet 22)	.57	.64	.66
Log A_v for 8 knots, curve x	2.69	2.69	2.69
s	.3814	.2686	.2476
R_d	128.7	108.9	105.8
I.H.P._d for $v_2=\text{I.H.P.}_d\times\left(\dfrac{\text{e.t.}}{E.T.}\right)^x$	2947	2766	2949
p.c. at 8 knots	.3393	.3616	.3392

Now by Sheet 20, the propulsive thrusts per square inch of projected area under Basic conditions, are for

P.A. ÷ D.A.	I.T.$_D$	P.C.	P.T.$_D$ = I.T.$_D$ × P.C.	P.T.$_p$ = P.T.$_D \div \frac{\text{P.A.}}{\text{D.A.}}$	1 − S	E.T.$_p$
.2	1.88	.709	1.233	6.16	.941	6.33
.3	3.74	.682	2.541	8.47	.941	9.00
.4	6.00	.619	3.714	9.29	.941	9.87

$$1-S=\frac{V\times 101.33}{P\times R}=\frac{\text{Speed Thrust}}{\text{Ind. Thrust}}=\frac{\text{Propulsive Thrust}}{\text{Effective Thrust}}.$$

Effective Thrust per square inch

$$\text{of projected area}=\text{E.T.}_p=\frac{\text{P.T.}_p}{1-\text{S}}.$$

Allowing an increase of 15 per cent on these effective thrusts before true cavitation begins, the values of e.t.$_p$ at true cavitation become

P.A. ÷ D.A.	E.T.$_p$ (Cav.)
.2	7.28
.3	10.35
.4	11.35

These final results show the necessity of large surface on propellers where large variation in speed with constant power is expected if a rapid falling off in propulsive efficiency at the low speeds is to be avoided.

While, without thrust deduction, the net effective (tow-rope) horse-power being delivered by the propellers is 1000, the actual gross work being performed by the screws on the water is that corresponding to the actual I.H.P.$_d$ = $K\times$I.H.P.$_p$, being expended on the propellers.

Thus it is seen that when the vessel is so loaded that eleven knots can be made with 1000 effective (tow-rope) horse-power, any one of the three propellers will answer, that of two blades being the most efficient, yet when the vessel is so loaded down either by her own cargo, by the condition of her bottom or by

P.A. ÷ D.A. =	.2	.3	.4
For 11 knots gross value $Z = \log \text{I.H.P.} - \log (K \times \text{I.H.P.}_p)$ =	.43363	.53563	.55763
For 8 knots gross value $Z = \log \text{I.H.P.} - \log \left(K \times \text{I.H.P.}_p \times \left(\frac{\text{e.t.}}{E.T.}\right)^2\right)$ =	.18943	.3419	.37711
For 11 knots gross e.h.p. ÷ E.H.P. (Sheet 21) =	.392	.306	.291
For 8 knots gross e.h.p. ÷ E.H.P. (Sheet 21) =	.662	.472	.437
For 11 knots gross e.h.p. =	1267	1267	1267
For 8 knots gross e.h.p. =	2141	1957	1901
Gross $e.t._p$ (11 knots) =	6.478	4.319	3.239
Gross $e.t._p$ (8 knots) =	10.95	6.669	4.859

weather conditions, or by having another vessel in tow, that this same effective horse-power will only deliver a speed of eight knots, the two-bladed propeller is entirely in adequate, as the total gross effective thrust per square inch of projected area is far in excess of that at the assumed cavitating point. In such a case, therefore, the three-bladed propeller might be chosen, as it is still within the limit for cavitation and has a considerable advantage in efficiency when running free over the four-bladed one. The four-bladed high-area propeller would, however, assure the smoothest running, but at the cost of higher power, would be well away from cavitation and would stand up better at still higher net thrusts.

The area may, in any case, be divided among four blades instead of three or two, except where blades would become exceedingly narrow without any particular loss in efficiency, as this latter is practically dependent upon the total projected area ratio and not upon the number of blades.

Attention should also be called to the change in revolutions at eight knots from those required at eleven knots, the effective horse-power remaining constant.

Attention must also be directed to the small influence of projected area ratio on revolutions for any given speed, where $v \div V$ corresponds approximately to "Upper E.T." limits, as the revolutions required at eleven knots for the two-bladed propeller of .2 projected area ratio are only 85.59, while those required by

the four-bladed wheel of .4 projected area ratio are 84.03, a decrease of only 2 per cent in revolutions for an increase of 100 per cent in surface, but this decrease in revolutions is accompanied by an increase of nearly 16 per cent in power.

Number of Blades Versus Projected Area Ratio in Determining Efficiency of Propulsion

As stated before, it is generally held that an increase in the number of blades of a propeller decreases its propulsive efficiency. This idea is held to be in error and that the propulsive efficiency depends practically upon the projected area ratio, as long as the blades are sufficiently narrow to escape interference with each other. There may, however, be a slight falling off in efficiency with the higher number of blades due to the greater number of blade edges around which eddying may occur.

To emphasize this point of efficiency depending mainly upon projected area ratio, the four-bladed propellers of four different vessels will be analyzed, each by reducing to $\frac{3}{4}$ its total projected area ratio for the Basic condition and then by using its full projected area ratio as the Basic condition for data, and comparing the results with those actually obtained on trial:

Vessel	1		2		3		4	
Slip Block Coef.	.805		.702		.79		.655	
No. of Propellers	1		1		1		2	
No. of blades	4	4	4	4	4	4	4	4
Total P.A. ÷ D.A.	.38	.38	.4	.4	.2828	.2828	.391	.391
$\frac{3}{4}$ P.A. ÷ D.A.	.285	....	.3	....	.2121	....	.293	
D	19'.5	19'.5	9'.67	9'.67	18'.5	18'.5	17'.25	17'.25
P	15'	15'	11'.5	11'.5	18'.75	18'.75	17'.8125	17'.8125
T.S.	6320	8200	6670	8600	4540	6270	6520	8400
R	103.2	133.9	219.6	283.7	78.12	107.9	120.3	155
$P \times R$	1548	2008	2525	3262	1465	2023	2143	2761
$1-S$	.945	.935	.92	.91	.942	.94	.91	.90
V	14.43	18.53	22.93	29.3	13.62	18.77	18.98	24.52
I.T._D	3.43	5.52	3.76	6.0	2.07	3.38	3.6	5.8
I.H.P.	7996	14442	3517	6247	4111	8019	18190	32660
P.C.	.633	.633	.62	.62	.691	.691	.626	.626
E.H.P.	5061	9142	2181	3873	2841	5541	11384	20440

Estimates of Performance

v_1	8.87		9.75		10		11.75	
$v \div V$	.614	.478	.425	.333	.734	.533	.619	.479
e.h.p.$_1$	970		218		1050 (Est.)		2295	
e.h.p.$_1$ ÷ E.H.P	.1917	.1061	.1	.05629	.3696	.1895	.2016	.1123
Z_1	.742	.994	1.0268	1.32	.459	.746	.712	.962
K	1.27	1.27	1.175	1.175	1.24	1.24	1.2	1.2
I.H.P.$_d$ = K × I.H.P.$_p$								
Estimated	1839	1860	389	351	1772	1785	4237	4278
Actual	1660	1660	408	408	1420	1420	3850	3850
log A_V	3.47	3.783	4.045	4.285	3.4	3.8	3.818	4.12
log A_{v1}	2.85	2.85	2.95	2.95	2.998	2.993	3.27	3.27
s_1 from actual power	.0476	.06403	.1155	.1271	.05056	.06735	.06728	.08345
Revs.								
Est. (Act. Power)	62.92	64.02	97.13	98.42	56.92	57.95	71.66	72.93
Actual	61.63	61.63	96	96	56	56	69	69
v_2	10.455		11.82		11.07		15.43	
$v \div V$	.724	.564	.515	.4	.813	.589	.813	.629
e.h.p.$_2$	1570		456		1370 (Est.)		5738	
e.h.p.$_2$ ÷ E.H.P	.3102	.1717	.2	.1178	.4822	.2473	.504	.2807
Z_2	.535	.781	.728	.96	.342	.628	.321	.571
K	1.27	1.27	1.175	1.175	1.24	1.24	1.2	1.2
I.H.P.$_d$ = K × I.H.P.$_p$								
Estimated	2963	3037	747	805	2319	2341	10423	10524
Actual	2755	2755	780	780	2200	2200	9950	9950
log A_V	3.47	3.783	4.045	4.285	3.4	3.8	3.818	4.12
log A_{v2}	3.05	3.05	3.21	3.21	3.13	3.13	3.55	3.55
s_2 from actual power	.0476	.06705	.1214	.1335	.0578	.07699	.09125	.1132
Revs.								
Est. (Act. Power)	74.16	75.71	118.6	120.2	63.5	64.82	96.59	98.98
Actual	73	73	117.8	117.8	65	65	94.5	94.5
v_3	11.12		12.65		12.3		18.1	
$v \div V$	.77	.600	.552	.432	.903	.655	.953	.738
e.h.p.$_3$	1885		545		2000 (Est.)		10328	
e.h.p.$_3$ ÷ E.H.P	.3724	.2062	.25	.1407	.704	.3609	.9072	.5053
Z_3	.457	.707	.63	.863	.167	.468	.048	.32
K	1.27	1.27	1.175	1.175	1.24	1.24	1.2	1.2
K × I.H.P.$_d$ Est.	3545	3601	970	1000	3470	3385	19540	18758
K × I.H.P.$_d$ Act.	3445	3445	990	990	3400	3400	19000	19000
log A_V	3.47	3.783	4.045	4.285	3.4	3.5	3.818	4.12
log A_{v3}	3.15	3.15	3.3	3.3	3.26	3.26	3.76	3.76
s_3	.04961	.0666	.1252	.1378	.06622	.08821	.1074	.1333
Revs. Est.	79.04	80.48	127.41	129.3	70.88	72.9	115.4	118.8
Revs. Act.	78.2	78.2	128	128	74.4	74.4	116	116

The closer agreement with actual powers and revolutions appears to rest with the method of reduction to the three-blade condition, for values of total P.A.÷D.A. above .3 and with the other below .3, but the agreement of the two methods throughout in the results obtained is very close, considering the approximations in data and in effective horse-power that must exist where the trials of the real vessel are taken into consideration. As the propulsive efficiency in both methods is taken as that of the full projected area ratio, without regard to the number of blades of the propeller, it would appear as if this latter had very little to do with the resultant efficiency.

Taylor, in commenting on the question of the relative efficiencies of two-, three-, and four-bladed propellers states, " There were tried a number of propellers with blades identical but differing in number—from two to six. It was found that efficiency was inversely as the number of blades; that is, a propeller with two blades was more efficient than a propeller with three identical blades, that one with three blades was more efficient than one with four identical blades and that one with four blades was more efficient than one with six identical blades."

" Also while total thrust and torque increase as number of blades is increased, the thrust and torque per blade fall off." . . . " It should be remembered that (this) refers to propellers working under identical conditions of slip, speed of advance, etc."

The pity of it is that Froude had covered identically the same ground and had arrived at the same conclusions, while neither Froude nor Taylor had attempted to keep the projected area constant and vary the number of blades. If they had done so the conclusion reached would have read as follows:

The number of blades of a propeller has no effect upon its propulsive efficiency provided each individual blade is sufficiently narrow throughout its length to insure against blade interference with the flow of the water through the propeller. Propulsive efficiency is based on projected area ratio of the propeller and that propeller having the greater projected area ratio will, as a general rule, have the lesser maximum propulsive efficiency, so long as the " Critical Thrusts " are not exceeded.

Estimates of Performance

In making estimates of expected performances of propellers in actual service, considerable differences between the estimated and the actual performances may be expected. These differences are caused by the following:

1. Conditions under which model of ship is tried and effective (tow-rope) horse-power obtained: Model wetted surface in the best of condition as to smoothness; water in tank smooth; air, still; model constrained to move in a perfectly straight course.

2. Conditions under which actual ship may be tried and the effects on performance:

(*a*) Wetted surface of hull may be more or less rough, producing increased resistance to motion through the water, produces increase in indicated or shaft horse-power and slight increase in revolutions for any given speed.

(*b*) Weather and sea conditions may be adverse,—same effect as (*a*).

(*c*) Strong following wind and sea,—opposite effect to (*a*) and (*b*).

(*d*) Form or trim of hull or adjustment of appendages be such as to cause a heavy wake,—this will increase the model tank effective (tow-rope) horse-power required for a given speed, but if the propeller is favorably located, a gain in propulsive efficiency due to wake gain will result;—the effect is to reduce power and revolutions for the given speed, as if the actual resistance of the hull at this speed had been reduced.

(*e*) Improper design of appendages producing excessive eddying of the water accompanied by a reduction in pressure in the locality in which the propeller operates;—this causes more or less increase in power over that estimated, while the revolutions of the propeller are considerably increased above those due to this power; this increase in revolutions and power being accompanied by more or less serious vibrations. The condition is abnormal but is frequently encountered.

(*f*) Erratic steering while on trial course,—causes apparent increase in power and revolutions for the noted speed.

(*g*) Errors of observations of speed.

(*h*) Errors of instruments for measurements of power and revolutions of the propelling engines.

This list of handicaps against which the estimator is pitted being very formidable, it becomes necessary, therefore, to be satisfied with any reasonably close estimate to the actual performance, particularly when the estimate of power exceeds the actual power necessary.

There is still one other source of error which may be caused by the propeller itself, and that is—

(*k*) Excessive roughness of propeller surfaces or excessive bluntness of edge,—these produce increased resistance per revolution, raise the power for a given speed but do not change the revolutions for the speed from what they would be if the blades were smooth and their edges fine.

Correction of Basic Propeller for Variation from Standard Form of Blade

Variations in blade form from the standard forms shown on Sheet 23, can be divided into three general classes, as follows:

1. Fan-shaped blades having the same total projected area as a standard form blade, whose diameter of propeller is greater than the diameter of the actual propeller, and whose blade projected area form coincides up to .7 Radius with the blade projected area form of the actual blade, the amount of surface cut off from the Basic blade by reducing the diameter to the actual diameter being restored by adding it in to the width of the blade between the .7 Radius of the Basic blade and the tip of the actual blade.

Such a blade is shown in Fig. 4.

With a propeller whose blade form has been so modified from that of the Basic propeller, the power and revolutions necessary to deliver a given effective horse-power at a given speed of vessel bear the following relations to these same quantities for the Basic propeller:

Let e.h.p. = Effective (tow-rope) horse-power of vessel.

v = speed of vessel corresponding to e.h.p.

I.H.P._{d_1} = Actual power required to deliver e.h.p. with the Basic propeller.

I.H.P._{d_2} = Actual power required to deliver e.h.p. with the actual propeller.

R_{d_1} = Revolutions corresponding to I.H.P._{d_1}, e.h.p. and v.

R_{d_2} = Revolutions corresponding to I.H.P._{d_2}, e.h.p. and v.

D_1 = Diameter, in feet, of Basic screw.

D_2 = Diameter, in feet, of actual screw.

Then, $$R_2 = R_1 \times \left(\frac{D_1}{D_2}\right)^{1/2},$$

$$\text{I.H.P.}_{d_2} = \text{I.H.P.}_{d_1} \times \frac{D_1}{D_2},$$

and the propulsive coefficients will vary directly as the diameters.

2. Oval blades having their greatest half cords of circular arc measurements of the projected area form at a radius greater than .7 Radius of the propeller, as shown in Fig. 5.

3. Oval blades having their greatest half cords of circular arc measurements of the projected area form at a radius less than .7 Radius of the propeller.

These two cases are just the opposite of each other and the corrections of power and efficiency for them are made in exactly the same manner, as follows:

Let the diameter of the circular arc of greatest projected area length

$= D_0$.

D = diameter of basic propeller.

D_1 = diameter of actual propeller.

e.h.p. = Effective horse-power delivered by Basic propeller.

v = Speed corresponding to e.h.p._1.

e.h.p._1 = Effective horse-power deliver by actual propeller at speed v.

$I.H.P._{\cdot d}$ = Indicated horse-power of Basic propeller to deliver e.h.p.$_1$ at speed v.

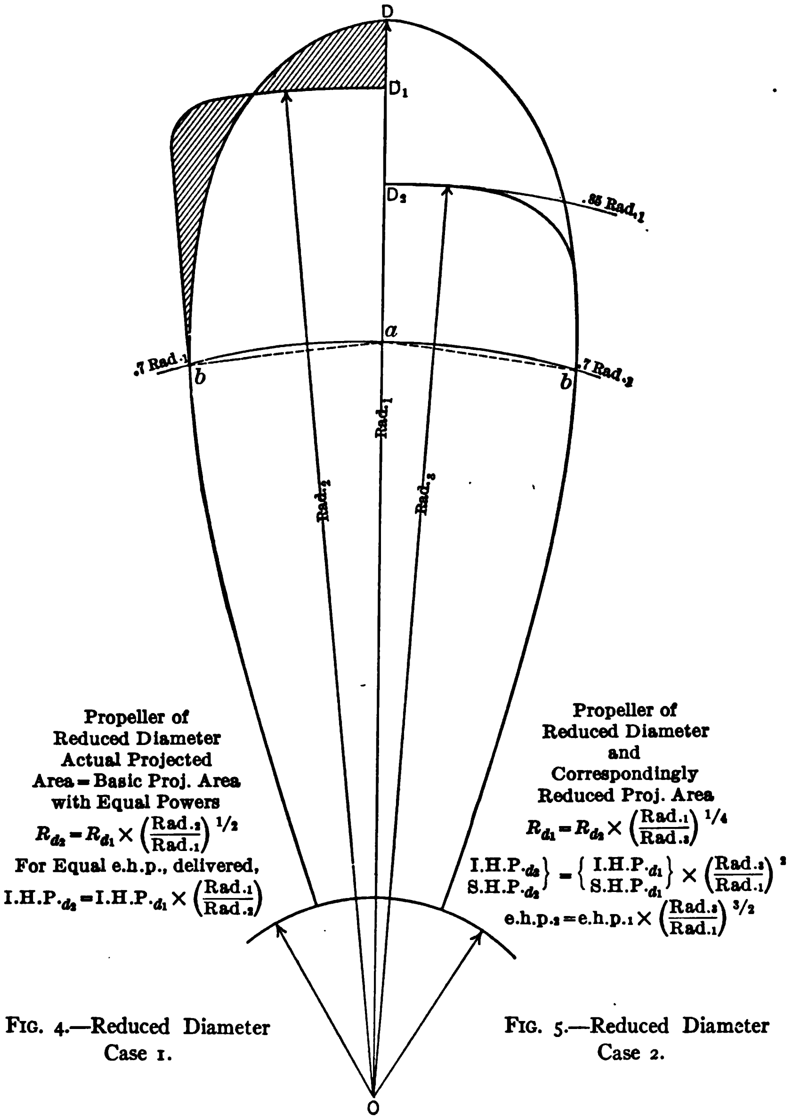


FIG. 4.—Reduced Diameter Case 1.

FIG. 5.—Reduced Diameter Case 2.

$I.H.P._{\cdot d_1}$ = Indicated horse-power of actual propeller to deliver e.h.p.$_1$, at speed v.

R_d and R_{d_1} = Revolutions of Basic and actual propellers, respectively, corresponding to the above conditions.

Then in the actual work of design, I.H.P.$_{d_1}$ and e.h.p.$_1$, and D_1 are replaced by

I.H.P.$_d$, e.h.p. and D, where

$$D = D_0 \div .7,$$

$$R_{d_1} = R_d \times \left(\frac{D}{D_1}\right)^{1/2},$$

$$\text{I.H.P.}_d = \text{I.H.P.}_{d_1} \times \left(\frac{D}{D_1}\right)^2,$$

$$\text{e.h.p.} = \text{e.h.p.}_1, \times \left(\frac{D}{D_1}\right)^{3/2}.$$

Should it be desired to analyze a given propeller, it becomes first necessary to obtain its Basic standard projected area form. This is readily done by taking the length of the cord of the half circular arc at diameter D_0 and dividing it by $\frac{D_0}{2}$, and with this quotient entering the table of half cords on Sheet 23, and from the column marked as .7 Radius, will be readily obtained the projected area ratio corresponding to this unit half cord length.

With propellers modified as in 1, 2 and 3, the modification apparently causes but slight change in the value of the thrust deduction factors from what these factors would be should the Basic propellers be used. 1 and 2 will probably cause a slight decrease and 3 an increase where the value of the thrust deduction varies with the relative tip clearance. These *slight* changes are due to the fact that the thrust deduction does not actually vary with the tip clearance except where blades of standard form are used. Where departures from this form exist it would be more correct to state that the " thrust deduction varies with the type of hull of vessel, location of propeller relative to the hull, and to the clearance between the center of pressure of the propeller blades and the hull."

Problems in Estimates of Performances

In the following problems the values of Z, used in estimating power and revolutions, are the empirical ones, the equation for Z not having been developed at the time this work was carried out. Also slight changes have been made in the curve of log A_V since this work was performed, particularly for values of v below ten knots:

Problem 1

Hull Data

Slip Block Coefficient = .60
Mean Referred Tip Clear. = 2'.88
Center of Propeller within limits of Load Water Plane
Propeller—Condition 1, Sheet 19
$K = 1.02$
Two Propellers

Basic Condition of Propeller

No. of Blades	3
P.A. ÷ D.A.	.328
D	18'.25
P	19'.75
T.S. for P.A. ÷ D.A. (Sheet 20)	7240
R = T.S. ÷ πD	126.67
$P \times R$	2494
$1-S$ for P.A. ÷ D.A. = .328 and Slip B.C. = .6 (Sheet 20) =	.8995
$V = P \times R \times (1-S) \div 101.33$	22.14
I.T.$_D$ for P.A. ÷ D.A. = .328 (Sheet 20)	4.35
2 I.H.P. = $2 \times (D^2 \times \text{I.T.}_D \times P \times R) \div 291.8$ =	24767
P.C. for P.A. ÷ D.A. = .328 (Sheet 20)	.665
E.H.P. = 2 × I.H.P. × P.C.	16420

e.h.p. ÷ E.H.P.	.075	.1	.2	.3	.4
e.h.p.	1232	1642	3284	4926	6568
v for e.h.p.	9.5	10.45	13.21	15.05	16.45
$v \div V$	.429	.472	.596	.679	.743
Z for e.h.p. ÷ E.H.P.	1.195	1.0268	.728	.5493	.4238
K	1.02	1.02	1.02	1.02	1.02
I.H.P.$_d = K$ I.H.P.$_p = K$ I.H.P. $\div 10^Z$ = Est. Power	1612	2373	4726	7131	9521
Actual Power	1600	2200	4800	7125	9300
Est. Revs. for v	52.8	58.63	74.05	84.5	92.68
Act. Revs. for v	52	57.5	73.6	84.5	92.5

e.h.p. ÷ E.H.P.	.5	.6	.7	.8
e.h.p.	8210	9852	11494	13136
v for e.h.p.	17.7	18.92	19.76	20.33
$v \div V$	.8	.854	.892	.919
Z for e.h.p. ÷ E.H.P.	.3267	.2432	.169	.1065
K	1.02	1.02	1.02	1.02
I.H.P.$_a$ = K I.H.P.$_p$ = K I.H.P. ÷ 10^Z = Est. Power	11909	14430	17119	19767
Actual Power	11600	14625	17200	19600
Est. Revs. for v	99.79	106.2	112	116
Act. Revs. for v	99.75	106.8	112.1	116.2

e.h.p. ÷ E.H.P.	.9	1.0	1.05	1.10
e.h.p.	14778	16420	17230	18062
v for e.h.p	20.8	21.24	21.44	21.62
$v \div V$	.939	.959	.968	.976
Z for e.h.p. ÷ E.H.P.	.0540	0.	.0225	.045
K	1.02	1.02	1.02	1.02
I.H.P.$_a$ = K I.H.P.$_p$ = K I.H.P. ÷ 10^Z = Est. Power	22307	25262	26605	28020
Actual Power	22200	25200	26600	28100
Est. Revs. for v	119.8	123.4	124.7	126.3
Act. Revs. for v	120.2	124.8	126.5	129.2

Problem 2

Hull Conditions

Slip B.C. = .627

Propeller located in Condition 2, Sheet 19

K = 1.22 (Lower Line of K)

Twin Propellers

In Condition 2, neglect tip clearance

Basic Condition of Propellers

Blades	3
P.A. ÷ D.A.	.304
D	15'.95
P	14'.436
T.S. for P.A. ÷ D.A. = .304 (Sheet 20)	6740
R = T.S. ÷ πD	134.4
$P \times R$	1943
$(1 - S)$ for P.A. ÷ D.A. = .304, Slip B.C. = .627 =	.904
$V = \{P \times R \times (1 - S)\} \div 101.33$	17.34
I.T$_D$ for P.A. ÷ D.A. = .304	3.85
2 × I.H.P.	13060
P.C.	.675
E.H.P.	8815
2 × S.H.P. = 2 × I.H.P. × .92	12020

ELECTRICALLY PROPELLED VESSEL

e.h.p. ÷ E.H.P.	.1	.2	.3	.4	.5
e.h.p.	882	1763	2645	3526	4408
v	8.77	11.18	12.77	14.0	14.95
$v \div V$	.506	.645	.736	.807	.862
Z	1.0268	.728	.5493	.4238	.3267
K	1.22	1.22	1.22	1.22	1.22
$\text{S.H.P.}_d = \text{S.H.P.}_p \times K$	1379	2743	4140	5527	6913
Act S.H.P_d. = Act. $K \times$ S.H.P._p	1500	2780	4160	5460	7000
Est. Revs	67.1	85.26	97.52	107.5	114.6
Actual Revs	67.3	86	98.2	108.3	116.2

Problem 3

Hull Conditions

4 Propellers
Slip Block Coef. = .62
Propellers located as in Condition 1, Sheet 19
Mean Referred Tip Clearance of Blades = 2′.24
$K = 1.12$

Basic Condition of Propellers

Blades	3
P.A. ÷ D.A.	.523
D	10′
P	8′.188
T.S.	11100
R	353.4
$P \times R$	2893
$1 - S$	.86
V	24.55
I.T._D	9.4
4 × I.H.P.	37192
P.C.	.54
E.H.P.	20084
S.H.P.	34296

e.h.p. ÷ E.H.P.	.1	.2	.3	.4
e.h.p.	2008	4017	6025	8034
v	11.1	13.85	15.75	17.2
$v \div V$	.411	.564	.641	.701
Z	1.0268	.728	.5493	.4238
K	1.12	1.12	1.12	1.12
$\text{S.H.P.}_d = K \times \text{S.H.P.}_p$	3641	7186	10843	14476
Act. S.H.P._d = Act. $K \times \text{S.H.P.}_p$	3800	7100	10500	14400
Est. Revs	158	197	226	249
Act. Revs	153	197.3	225.9	247.1

e.h.p. ÷ E.H.P.	.5	.6	.7	.8
e.h.p.	10042	12050	14059	16067
v	18.45	19.55	20.35	20.95
$v \div V$	.751	.796	.829	.853
Z	.3267	.2432	.169	.1065
K	1.12	1.12	1.12	1.12
S.H.P.$_{\cdot d}$ = $K \times$ S.H.P.$_{\cdot p}$	18103	21941	26929	30055
Act. S.H.P.$_{\cdot d}$ = Act. $K \times$ S.H.P.$_{\cdot p}$	18900	22300	25300	28800
Est. Revs	271	287.5	300	311.5
Act. Revs	266.9	285.5	298.9	312.2

Problem 4

Hull Conditions

Slip B.C. = .662

Propellers located in Condition 3, Sheet 19

Neglect Tip Clearance

$K = 1.07$

Two Propellers

Basic Conditions of Propellers

Blades	3
P.A. ÷ D.A.	.315
D	17′.54
P	18′
T.S.	6940
R	126
$P \times R$	2267
$1 - S$	.903
V	20.2
I.T.$_{\cdot D}$	4.03
2 I.H.P.	19265
P.C.	.674
E.H.P.	12985

e.h.p. ÷ E.H.P.	.1	.2	.3	.4	.5
e.h.p.	1299	2597	3896	5194	6493
v	10.4	12.96	14.58	15.55	16.75
$v \div V$	.515	.641	.722	.77	.829
Z	1.0268	.728	.5493	.4238	.3267
K	1.07	1.07	1.07	1.07	1.07
I.H.P.$_{\cdot d}$ = $K \times$ I.H.P.$_{\cdot p}$	1938	3856	5819	7769	9715
Act. I.H.P.$_{\cdot d}$ = Act. $K \times$ I.H.P.$_{\cdot p}$	1900	4260	6150	7700	9900
Est. Revs	63.02	78.63	88.76	95.5	103.1
Act. Revs	62.6	77.4	89	95.8	104.2

e.h.p. ÷ E.H.P.	.6	.7	.8	.9
e.h.p.	7771	9090	10388	11687
v	17.7	18.4	18.97	19.48
$v \div V$	.876	.911	.939	.964
Z	.2432	.169	.1065	.05402
K	1.07	1.07	1.07	1.07
I.H.P.$_d$ = K × I.H.P.$_p$	11774	13968	16130	18202
Act. I.H.P.$_d$ = Act. K × I.H.P.$_p$	12150	14350	16150	18200
Est. Revs	109.2	114.2	118.3	122.2
Act. Revs	111.2	116.6	121.8	124.9

Problem 5

Hull Conditions

Slip B.C. = .62
Propeller located in Condition 1, Sheet 19
Mean Tip Clearance = 3′.5
K = 1

Basic Conditions of Propellers

Blades	3
P.A. ÷ D.A.	.501
D	9′.583
P	8′.193
T.S.	10580
R	351.2
$P \times R$	2878
$1 - S$	.87
V	24.71
I.T.$_D$	8.75
4 × I.H.P.	31732
P.C.	.554
E.H.P.	17580
4 × S.H.P.	29196

e.h.p. ÷ E.H.P.	.1	.2	.3	.4
e.h.p.	1758	3516	5274	7032
v	10.31	12.96	14.81	16.3
$v \div V$	.417	.524	.6	.66
Z	1.0268	.728	.5493	.4238
K	1	1	1	1
S.H.P.$_d$ = K × S.H.P.$_p$	2745	5461	8241	11002
Act. S.H.P.$_d$ = Act. K × S.H.P.$_p$	2750	5400	8100	11050
Est. Revs	148.8	190.2	217.2	239.6
Act. Revs	153	189.5	218	241

e.h.p. ÷ E.H.P	.5	.6	.7	.8
e.h.p.	8790	10548	12306	14064
v	17.44	18.42	19.32	20.1
$v \div V$	.706	.745	.782	.813
Z	.3267	.2432	.169	.1065
K	1	1	1	1
$S.H.P._d = K \times S.H.P._p$	13759	16676	19782	22844
Act. $S.H.P._d$ = Act. $K \times S.H.P._p$	13700	16300	19150	22000
Est. Revs	257.2	273.2	288.2	303.5
Act. Revs	259	274.5	288.5	302.5

e.h.p. ÷ E.H.P	.9	1.0	1.05
e.h.p.	15822	17580	18459
v	20.7	21.13	21.3
$v \div V$	.837	.855	.862
Z	.05402	0	.0225
K	1	1	1
$S.H.P._d = K \times S.H.P._p$	25779	29196	30745
Act. $S.H.P._d$ = Act. $K \times S.H.P._p$	25300	28300	30000
Est. Revs	314.6	327.3	331.1
Act. Revs	315	326	331.5

Problem 6

Hull Conditions

Slip B.C. = .61
No. of Propellers = 4
Propellers located in Condition 1, Sheet 19
Mean T.C. = 3′.33
$K = 1$

Basic Conditions of Propellers

Blades	3
P.A. ÷ D.A.	.558
D	9′.17
P	8′.5
T.S.	12080
R	419.7
$P \times R$	3568
$1 - S$	.85
V	29.93
$I.T._D$	10.6
4 × I.H.P.	43596
P.C.	.527
E.H.P	22976
4 × S.H.P.	40108

e.h.p. ÷ E.H.P.	.1	.2	.3	.4
e.h.p.	2298	4595	6893	9190
v	12.1	15.05	17	18.8
$v \div V$	.404	.503	.568	.629
Z	1.0268	.728	.5493	.4238
K	1	1	1	1
S.H.P.$_d = K \times$ S.H.P.$_p$	3771	7503	11329	15116
Act. S.H.P.$_d$ = Act. $K \times$ S.H.P.$_p$	3900	7800	11600	15600
Est. Revs	172.1	215.9	246.1	272.9
Act. Revs	173.5	219	247.5	273.5

e.h.p. ÷ E.H.P.	.5	.6	.7	.8
e.h.p.	11488	13786	16083	18380
v	19.81	20.51	21.08	21.6
$v \div V$	.662	.685	.704	.722
Z	.3267	.2432	.169	.1065
K	1	1	1	1
S.H.P.$_d = K \times$ S.H.P.$_p$	18903	22910	27100	31395
Act. S.H.P.$_d$ = Act. $K \times$ S.H.P.$_p$	19350	22900	27000	31700
Est. Revs	291.7	311.6	326.8	341.1
Act. Revs	292.5	309	325	335

Problem 7

In this problem the vessel was a two-shaft destroyer with the propellers located well aft abreast the stern post. The vessel squatted heavily at high speeds. The squat begins approximately at $v \div \sqrt{\text{L.L.W.L.}} = 1.48$ and is fully accomplished at $v \div \sqrt{\text{L.L.W.L.}} = 2.13$, the value of log A_s in the apparent slip equation passing slowly during the process from Curve X to Curve Y on Sheet 21, on a straight line tangent to Y at the point of accomplishment.

In the case in question, the propeller blades were not of standard form, being of oval form but having the greatest circular width of projection at a distance out from the center corresponding to .7 of 7′.3 diameter, .7 D being the diameter of the estimated center of pressure of the standard blade form. The cord of the half arc at this point, divided by $\frac{7.3}{2}$, corresponds to the dimension of this cord given in the table on Sheet 25, for a projected area ratio of .617, and the Basic propeller for analysis is therefore taken as having a pitch = pitch of actual propeller, diameter = 7′.3 and projected area ratio = .617, while the Basic S.H.P., I.H.P. and E.H.P. of the actual propeller are taken as those of the basic propeller $\times \left(\frac{D_1}{D_2}\right)^2 = \left(\frac{6.67}{7.3}\right)^2$.

Hull Conditions

Slip B.C. = .341
$K = 1$
Two Screws
L.L.W.L. = 285′

$$\frac{v}{\sqrt{285}} = 1.48$$

$v = 24.99$ Squat begins

$$\frac{v}{\sqrt{285}} = 2.13$$

$v = 35.96$ Squat accomplished

Basic Conditions of Propellers

	Actual	Basic
Blades	3	3
P.A. ÷ D.A.	.587	.617
D	6′.67	7′.3
P	6′.17	6′.17
T.S.	....	14250
R	....	632.3
$P \times R$	....	3901
$1 - S$	....	817
V	....	31.45
I.T.$_D$	....	12.55
2 I.H.P.	$14930 = \left(\frac{6.67}{7.3}\right)^2 \times$	17882
P.C	....	.5225
E.H.P	7838	
2 S.H.P	13734	

e.h.p. ÷ E.H.P	.025	.05	.075	.1
e.h.p.	196	392	588	784
v	10.25	12.9	14.70	16.10
$v \div V$	.334	.422	.477	.522
(Est. Curve) Z	1.668	1.355	1.1715	1.0414
K	1	1	1	1
S.H.P.$_a$ = $K \times$ S.H.P.$_p$	320	660	1060	1358
Act. S.H.P.$_a$ = Act. $K \times$ S.H.P.$_p$	325	780	1150	1500
Est. Revs	182.5	231.2	264.5	290
Act. Revs	182.5	230	262	289

e.h.p. ÷ E.H.P	.2	.3	.4	.5
e.h.p.	1568	2352	3136	3919
v	20.2	22.25	23.85	25.38
$v \div V$	.649	.701	.776	.823
(Est. Curve) Z	.7279	.5445	.4144	.3135
K	1	1	1	1
S.H.P.$_a$ = $K \times$ S.H.P.$_p$	2794	4262	5750	7254
Act. S.H.P.$_a$ = Act. $K \times$ S.H.P.$_p$	2900	4250	5500	7250
Est. Revs	366.7	411	448.6	488.9
Act. Revs	367	411	449	489

e.h.p. ÷ E.H.P.	.6	.7	.8	.9
e.h.p.	4703	5287	6271	7054
v	26.77	27.65	29.12	30.33
$v \div V$	.867	.905	.942	.978
(Est. Curve) Z	.2310	.161	.1009	.04770
K	1	1	1	1
$S.H.P._a = K \times S.H.P._p$	8770	10304	11834	13376
Act. $S.H.P._a$ Act. $K \times S.H.P._p$	8950	9950	11500	12950
Est. Revs	530.2	555.9	604.1	642.6
Act. Revs	528	555	605	648

(This problem is uncorrected for increase in efficiency due to cutting off of blade tips, as given by equations under Case 2, change in blade form.)

Problem 8

While Problem 7 was an example of change in blade form due to cutting of the tips of the Basic propeller with consequent reduction in projected area as described in Case 2, change of blade form, the present problem is one coming under Case 1, where the blade tips are cut off but the total Basic projected area is retained by broadening the ends of the blades outside the center of pressure.

The vessel was of the well-known naval collier type, fan-tailed stern, twin screw, with the propellers located as in Position 2, Sheet 19. The slip block coefficient is .665. The maximum diameter of propeller that could be carried was 16 ft. 6 in. The tank curve of e.h.p. with all appendages was available for use in the estimate.

From Sheet 19, the value of K for this S.B.C. and location of propeller $= 1.22$.

BASIC CONDITIONS OF PROPELLER

Propeller	Basic	Actual
D	17'.75	16'.5
P	16'.42	16'.42
$\frac{P.A.}{D.A.}$ (3-bladed)	.32	.37
T.S.	7050	
$P \times R$	2076	
S.B.C.	.665	
$1-S$	.91	
V	18.64	
$I.T._D$	4.16	
I.H.P.	9324	
P.C.	.67	
E.H.P	6247	

ESTIMATED AND ACTUAL PERFORMANCES

v	e.h.p.	$\frac{\text{e.h.p.}}{\text{E.H.P.}}$	Z	I.H.P.$_p$	K	I.H.P.$_d$	$\frac{17.75}{16.5}$	I.H.P.$_{d1}$
9	475	.076	1.15	660	1.22	805	1.076	866
10	640	.1024	1.02	870	1.22	1086	1.076	1168
11	845	.1353	.905	1160	1.22	1416	1.076	1524
12	1100	.1762	.795	1495	1.22	1824	1.076	1964
13	1400	.2241	.670	1994	1.22	2432	1.076	2617
14	1650	.2641	.60	2342	1.22	2857	1.076	3074
15	2250	.3602	.46	3233	1.22	3944	1.076	4244

v	Actual I.H.P.$_{d1}$	Log Av	Log A_s	s	Basic R_d	$\left(\frac{D}{D_1}\right)^{\frac{1}{5}}$	Revs. Est.	Revs. Act.
9	875	3.8	2.91	.0605	59.12	1.037	61.3	61.6
10	1168	3.8	3.02	.0633	65.88	1.037	68.3	68.3
11	1512	3.8	3.12	.0654	72.63	1.037	75.3	75.1
12	1938	3.8	3.23	0654	79.24	1.037	82.2	82
13	2450	3.8	3.335	.0685	86.12	1.037	89.3	88.8
14	3109	3.8	3.43	.0647	92.37	1.037	95.8	96.2
15	3900	3.8	3.52	.0725	99.81	1.037	103.5	103.8

The quantities in the estimate are for one propeller only and the powers should be doubled for the total.

All of the foregoing propellers were three-bladed, of manganese bronze, machined to pitch, the edges sharpened and the blades highly polished.

Turning now to the four-bladed propellers, none of those in the following problems were more than simply smoothed off, there being no machining to insure pitch and no particular care taken to sharpen blade edges.

Problems in Estimates of Performances of 4-bladed Propellers

Problem 9

Hull Conditions	*Basic Conditions of Propeller*	
Slip B.C. = .80	Blades	4
Single Screw	$\frac{4}{3}$ P.A. ÷ D.A. = Total Proj. Area Ratio	.38
Draught greater than 20 ft.	P.A. ÷ D.A.	.285
	D	19'.5
	P	15'
	T.S. for P.A. ÷ D.A.	6320
	R	103.2
	$P \times R$	1548
	$1-S$	.945
	V	14.43
	I.T.$_D$	3.43
	$\text{I.H.P} = \dfrac{D^2 \times \text{I.T.}_D \times P \times R}{252.41}$	7996
	P.C. for $\frac{4}{3}$ P.A. ÷ D.A.	.633
	E.H.P.	5061

e.h.p.	970	1570	1885	Notes: Low speed obtained by two runs only, one with and one against the tide. Shallow water course with heavy tide effect.
e.h.p. ÷ E.H.P.	.1917	.3102	.3724	
v	8.87	10.455	11.12	
$v \div V$	.614	.724	.77	
Z	.742	.535	.457	
K	1.27	1.27	1.27	
I.H.P.$_d$ = $K \times$ I.H.P.$_p$	1839	2963	3545	
Act. I.H.P.$_d$ = Act. $K \times$ I.H.P.$_p$	1660	2755	3445	
Est. Revs	63.42	74.63	79.15	
Act. Revs	61.63	73	78.2	

Problem 10

Hull Conditions	*Basic Conditions of Propeller*	
Single Screw	Blades	4
Slip B.C. = .702	$\frac{4}{3}$ (P.A. ÷ D.A.)	.4
Draught = 12 ft.	P.A. ÷ D.A.	.3
$K = 1.175$	D	9′.67
	P	11′.5
	T.S.	6670
	R	219.6
	$P \times R$	2525
	$1 - S$	.92
	V	22.93
	I.T.$_D$	3.76
	I.H.P.	3517
	P.C.	.62
	E.H.P.	2181

e.h.p.	218	456	545
e.h.p. ÷ E.H.P.	.1	.4822	.704
v	9.75	11.82	12.65
$v \div V$	.425	.515	.552
Z	1.0268	.728	.63
K	1.175	1.175	1.175
I.H.P.$_d$ = $K \times$ I.H.P.$_p$	389	747	970
Act. I.H.P.$_d$ = Act. $K \times$ I.H.P.$_p$	408	780	990
Est. Revs.	96.53	118.4	127
Act. Revs.	96	117.8	128

Problem 11

Hull Conditions	*Basic Conditions of Propellers*	
Single Screw	Blades	4
Slip B.C. = .79	$\frac{4}{3}$ (P.A. ÷ D.A.)	.2828
Draught greater than 20 ft.	P.A. ÷ D.A.	.2121
$K = 1.24$	D	18′.5
	P	18′.75
	T.S.	4540
	R	78.12
	$P \times R$	1465
	$1 - S$	.942
	V	13.62
	I.T.$_D$	2.07
	I.H.P.	4111
	P.C.	.691
	E.H.P.	2841

Est. e.h.p.	1050	1125	1200	1290
e.h.p. ÷ E.H.P.	.3696	.396	.4224	.4541
v	10	10.27	10.55	10.82
$v \div V$	.718	.737	.757	.777
Z	.46	.427	.4	.37
K	1.24	1.24	1.24	1.24
I.H.P.$_d$ = $K \times$ I.H.P.$_p$	1768	1906	2030	2175
Act. I.H.P.$_d$ = Act. $K \times$ I.H.P.$_p$	1420	1600	1800	2000
Est. Revs.	57.73	59.33	60.92	62.38
Act. Revs.	56	58.7	61.4	63.65

Est. e.h.p.	1370	1500	1600	1725
e.h.p. ÷ E.H.P.	.4823	.5281	.5633	.6073
v	11.07	11.31	11.54	11.76
$v \div V$	.795	.812	.83	.845
Z	.342	.302	.276	.24
K	1.24	1.24	1.24	1.24
I.H.P.$_d$ = $K \times$ I.H.P.$_p$	2319	2543	2700	2933
Act. I.H.P.$_d$ = Act. $K \times$ I.H.P.$_p$	2200	2400	2600	2800
Est. Revs.	63.81	65.39	66.68	67.92
Act. Revs.	65	67.1	68.65	70.15

Est. e.h.p.	1820	1910	2000
e.h.p. ÷ E.H.P.	.6407	.6724	.7041
v	11.98	12.18	12.36
$v \div V$	.861	.875	.888
Z	.214	.19	.17
K	1.24	1.24	1.24
I.H.P.$_d$ = $K \times$ I.H.P.$_p$	3114	3291	3447
Act. I.H.P.$_d$ = Act. $K \times$ I.H.P.$_p$	3000	3200	3400
Est. Revs.	69.36	70.5	71.49
Act. Revs.	71.65	72.9	74.4

The e.h.p.'s of the foregoing vessel were estimated, and the trials were held over a shallow water course with heavy tidal currents, one run in each direction being made for each point.

Problem 12

Hull Conditions	*Basic Conditions of Propellers*	
Twin Screw	Blades	4
Slip B.C. = .655	$\frac{4}{3}$ (P.A. ÷ D.A.)	.391
Propellers located in Position 2, Sheet 19	P.A. ÷ D.A	.293
$K = 1.2$	D	17′.25
	P	17′.8125
	T.S.	6520
	R	120.3
	$P \times R$	2143
	$1 - S$	.91
	V	18.98
	I.T.$_D$	3.6
	I.H.P.	18190
	P.C.	.636
	E.H.P	11384

e.h.p. ÷ E.H.P.	.1008	.2016	.3025	.4032	.5041
e.h.p.	1148	2295	3443	4590	5738
v	9.4	11.75	13.4	14.6	15.43
$v \div V$	.495	.619	.706	.77	.813
Z	1.023	.719	.542	.42	.322
K	1.2	1.2	1.2	1.2	1.2
I.H.P.$_d$ = $K \times$ I.H.P.$_p$	2070	4169	6266	8299	10400
Act. I.H.P.$_d$ = Act. $K \times$ I.H.P.$_p$	2000	3850	5940	8080	9950
Est. Revs	58.33	73.97	83.53	91.25	97.08
Act. Revs	54.5	69	80	88.4	94.5

e.h.p. ÷ E.H.P.	.6049	.7057	.8065	.9075
e.h.p.	6886	8033	9181	10328
v	16.15	16.83	17.52	18.1
$v \div V$	.851	.886	.923	.954
Z	.24	.168	.101	.043
K	1.2	1.2	1.2	1.2
I.H.P.$_d$ = $K \times$ I.H.P.$_p$	12561	14826	17299	19770
Act. I.H.P.$_d$ = Act. $K \times$ I.H.P.$_p$	11850	14200	16750	19000
Est. Revs	102.04	106.8	111.7	115.7
Act. Revs	100	105.75	111.5	116

There is evidently considerable wake gain with this hull at speeds below seventeen knots, reducing power and revolutions. This excess wake is clearly indicated by the e.h.p. curve.

Problems Showing Effect of Varying Conditions

Problem 13. Smooth Versus Fair Condition of Ship's Bottom

Hull Condition	*Basic Conditions of Propellers*	
Twin Screws	Blades	3
Slip B.C. = .608	P.A. ÷ D.A.	.32
Propellers located in Condition 1, Sheet 19	D	18′.65
$K = 1$ for smooth bottom	P	19′.99
	T.S.	7110
	R	121.4
	$P \times R$	2426
	$1 - S$	.899
	V	21.52
	I.T.$_D$	4.17
	2 I.H.P	24115
	P.C.	.671
	E.H.P	16181

The ship with the smooth bottom was just out of dry dock when tried, while the sister ship had been out of dock a few weeks, just sufficient to destroy the smooth polished surface of the bottom paint. The effect upon the performance will be seen to be very pronounced.

e.h.p. ÷ E.H.P	.1	.2	.3	.4
e.h.p.	1618	3236	4854	6472
v	9.55	12.35	14.15	15.51
$v \div V$	.443	.574	.657	.720
Z	1.0268	.728	.5493	.4238
K	1	1	1	1
I.H.P.$_d$ = $K \times$ I.H.P.$_p$	2267	4511	6807	9089
Act. I.H.P.$_d$ = Act. $K \times$ I.H.P.$_p$ (Smooth)	2200	4450	6600	8600
Act. I.H.P.$_d$ = Act. $K \times$ I.H.P.$_p$ (Rough)	2400	4900	7300	9800
$K \times$ I.H.P.$_d$ (Rough) ÷ $K \times$ I.H.P.$_d$ (Smooth)	1.09	1.10	1.10	1.14
Est. Revs. (Smooth)	54.33	68.95	79.2	88.36
Act. Revs. (Smooth)	52	69	79	87.4
Act. Revs., (Rough)	54.5	70	80.5	88.7

e.h.p.÷E.H.P.	.5	.6	.7	.8
e.h.p.	8090	9709	11327	12945
v	16.7	17.7	18.61	19.42
$v+V$	.776	.822	.865	.902
Z	.3267	.2432	.169	.1065
K	1	1	1	1
I.H.P.$_d$=K×I.H.P.$_p$	11366	13775	16343	18871
Act. I.H.P.$_d$=Act. K×I.H.P.$_p$ (Smooth)	11100	13450	15900	18450
Act. I.H.P.$_d$=Act. K×I.H.P.$_p$ (Rough)	12300	14900	17350	20000
K×I.H.P.$_d$ (Rough)÷K×I.H.P.$_a$ (Smooth)	1.108	1.108	1.09	1.084
Est. Revs. (Smooth)	94.01	99.73	105.5	110.5
Act. Revs. (Smooth)	95.1	101.5	106.2	111
Act. Revs. (Rough)	96.2	102.2	107.2	112

e.h.p.+E.H.P.	.9	1.0	1.05	1.10
e.h.p.	14563	16181	16990	17799
v	20.08	20.6	20.81	21.04
$v+V$	.933	.957	.967	.977
Z	.05402	0	.0225	.045
K	1	1	1	1
I.H.P.$_d$=K×I.H.P.$_p$	21295	24115	25397	26748
Act. I.H.P.$_d$=Act. K×I.H.P.$_p$ (Smooth)	20900	23200	24500	26200
Act. I.H.P.$_d$=Act. K×I.H.P.$_p$ (Rough)	22700	25500	27200	28400
K×I.H.P.$_d$ (Rough)÷K×I.H.P.$_d$ (Smooth)	1.086	1.10	1.11	1.084
Est. Revs. (Smooth)	114.3	118.1	119.4	121.3
Act. Revs. (Smooth)	115.8	120	121.7	123.9
Act. Revs. (Rough)	116.3	120.2	122.2	124.4

The rough-bottomed vessel is shown by this table to have required an average of 10 per cent higher power for the same speeds than the smooth-bottomed one, and this even when the bottom was in such condition as to be rated "clean."

Problem 14

In this problem is given the case of a vessel whose bottom was reported clean and in too good condition to justify docking before trial. The bottom was painted with a grade of paint that even when newly applied was rough and scaly. That this condition of paint had an extremely malign influence was evidenced by the fact that several months after the acceptance

trial of the vessel, a service trial was reported on which the acceptance trial was practically duplicated, the bottom being reported clean, yet the log bore the entry "217 days out of dock."

The vessel was fitted with a single propeller driven by reduction gears, the power delivered to the gears being measured forward of the gears.

In the estimate of performance, a loss of $1\frac{1}{2}$ per cent of power has been allowed through the gears and $2\frac{1}{2}$ per cent through the thrust bearing, making a total loss from the measured shaft horse-power of 4 per cent, in addition to the loss by thrust deduction. This latter loss, only, is taken account of in the estimate of revolutions.

The problem is to estimate the actual effective (tow-rope) horse-power delivered by the propeller and by comparison with the model tank curve of effective horse-power find the increase in resistance due to the roughness of the ship's bottom.

Hull Conditions

Single Screw
Slip B.C. = .66
Draught 20 ft.
$K = 1.07$

Basic Condition of Propeller

Blades	4
$\frac{4}{3}$ (P.A. ÷ D.A.)	.327
P.A. ÷ DA	.246
D	15′.5
P	16′
T.S.	5400
R	110.9
$P \times R$	1774
$1-S$	.913
V	15.99
I.T.$_D$	2.67
I.H.P.	4509
P.C.	.667
E.H.P.	3008
S.H.P.	4148

Total S.H.P.$_d$ = Total S.H.P.$_p(K+K^1)$	520	1068	1635	2200
K	1.07	1.07	1.07	1.07
K^1	.04	.04	.04	.04
S.H.P.$_p$	468	962	1473	1982
Z = log S.H.P. − log S.H.P.$_p$	.948	.635	.450	.321
e.h.p. ÷ E.H.P.	.117	.248	.38	.525
① e.h.p. (Trial)	352	746	1143	1579
② e.h.p. (Model)	301	602	903	1204
① ÷ ②	1.17	1.239	1.266	1.311
v	7.5	9.8	11.3	12.4
$v \div V$	.47	.613	.706	.775
Est. Revs	50.79	68.36	78.91	86.57
Act. Revs	51	67.5	78	86

Total S.H.P.$_a$ = Total S.H.P.$_p(K+K^1)$....	2770	3317	3846	4341
K..................................	1.07	1.07	1.07	1.07
K^1.................................	.04	.04	.04	.04
S.H.P.$_p$..............................	2495	2988	3465	3911
Z = log S.H.P. − log. S.H.P.$_p$............	.221	.142	.078	.026
e.h.p. ÷ E.H.P........................	.641	.745	.86	.955
① e.h.p. (Trial).........................	1928	2241	2587	2873
② e.h.p. (Model).......................	1505	1806	2107	2408
① ÷ ②..................................	1.28	1.24	1:228	1.193
v....................................	13.3	14.3	14.75	15.45
$v \div V$.................................	.832	.894	.922	.966
Est. Revs............................	93.08	99.81	103.6	108.3
Act. Revs............................	92.6	98.2	103.5	108.5

It will be noted that at the low speeds where the resistance is mainly frictional, the effect of the roughness of bottom on the performance is a maximum.

Wake Gain

In the foregoing examples, particularly those in which the maximum speed of vessel was high, there will be remarked generally an excess of estimated power over the actual power at certain speeds. This difference can be attributed to what is commonly called "wake gain," and is especially prominent in cases where the value of K is low.

In Taylor's "Speed and Power of Ships" is shown a diagram of the humps and hollows occurring in the resistance curves of ships (Fig. 2). The locations of these humps and hollows depend upon the load water line length of the ship and upon the speed, but always occur at about the same values of $v \div \sqrt{\text{L.L.W.L.}}$

The humps are caused by increases above the normal wake of the hull, while the hollows are caused by the wake drawing down towards the normal wake and in some cases falling below it, even in certain instances falling so far below it as to become negative.

These humps, in the model tank, appear as abnormal increases in resistance and the hollows as corresponding decreases.

When, however, the vessel is propelled by its own power, with

its screw propellers working in these wakes, these abnormally high wakes add to the thrust of the propeller per revolution, the power per revolution remaining constant, so that the propulsive coefficients realized at these positions of abnormally high wake become themselves abnormal. As the hollows fall to the normal condition, the propulsive coefficients likewise become normal, and finally when the actual wake falls below the normal wake the propulsive coefficients fall below the normal ones until wher the wake has actually become negative the revolutions become unduly high and this undue increase in revolutions will be accompanied by an excessive increase in power.

In vessels having the propellers located as shown in Condition 1, Sheet 18, the benefit of this wake gain may, however, be completely lost by locating the propellers with insufficient tip clearance between the propeller blades and the hull. With propellers so located, there exists with such hulls a current of high aft-flow velocity close in to the skin of the ship, and if the tip clearance of the propeller be insufficient, the tips of the blades will penetrate into this high velocity layer, and due to the low-pitch angle of the propellers at the tips, the thrust per revolution will be decreased and the velocity of flow of this malign current will be retarded intermittently as the tip of each blade enters it. The writer has freshly in mind the case of a destroyer, where with propellers 8 ft. in diameter a violent pulsation of the ship's bottom occurred at high speeds, at a location 30 ft. forward of the propellers. Upon fitting another set of propellers having the same pitch and projected area but with a diameter of only 7 ft. 6 in., the pulsation completely disappeared. Apparently the additional 6 in. in diameter was sufficient to cut into this current of high velocity of flow and produce an action similar in all respects to that of the well-known "water hammer."

Undoubtedly, some of the water in the rapid flowing skin current will be thrown off radially by the propeller blades by this periodic checking of flow, and when the tips of the propellers pass in close proximity to the hull the water thus thrown off inpinges violently on the hull plates, produces violent local vibrations which may be of such intensity as to break in the hull

plating, and at the same time produces a loss in power which may, and usually does, offset the possible "wake gain."

In high-speed vessels of normal form, such as torpedo boats and destroyers, and in vessels having the propellers located as in Condition 1, Sheet 18, it is recommended that the relative referred tip clearance of the propellers be not less than 3 ft., and in excess of that, if possible.

In cases where large tip clearances are provided, a "wake gain" may confidently be expected and, if desired, may be allowed for in making the preliminary estimates of performance, but where the tip clearance is small it is better to neglect the wake gain as it may be more than offset by the exaggerated thrust deduction occurring.

Correction of Effective Horse-power Curve for Expected Wake Gain

In some cases abnormal increases and decreases in wake, from the wake usually encountered with the fullness of hull of the vessel under consideration, are clearly indicated by the characteristics of the curve of effective horse-power (tow-rope horse-power), obtained by towing the model of the vessel in the model tank. This is clearly shown by the curves given in Figs. 6 and 7.

These curves are curves of effective horse-power (tow-rope horse-power), per ton of displacement, erected on speed divided by the square root of the length on the load water line of the vessel under consideration, that is, the ordinates are $\frac{\text{e.h.p.}}{\text{Tons Disp.}}$, while the abscissas are $v \div \sqrt{\text{L.L.W.L.}}$

On both of these curves occur humps, more or less distinctly shown, which indicate an increased wake over that occurring throughout the major portions of the curves, while in Fig. 7 is shown the characteristic change in the curve which occurs when a rapid decrease in wake is encountered.

In cases where these humps occur it is perfectly safe practice to take a spline and follow the main character of the curve, ignoring the humps, in making the estimate of speeds, but in

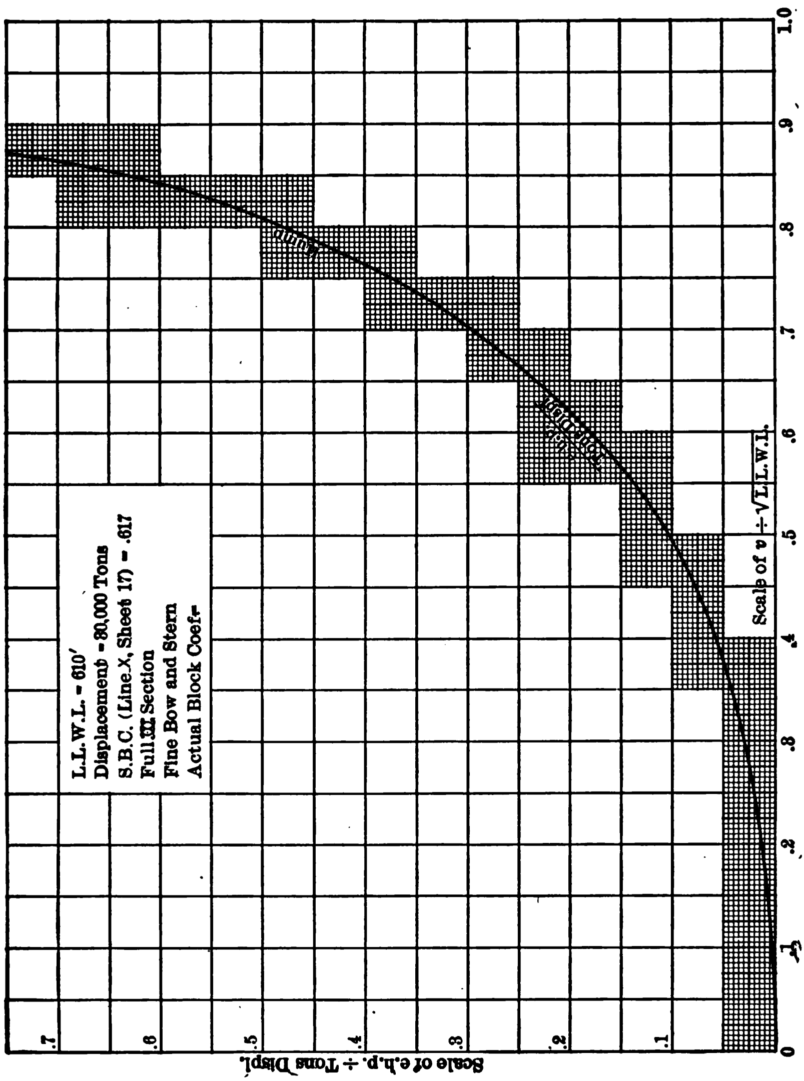


FIG. 6.—The Resistance Curve Showing Hump Caused by Abnormal Wake.

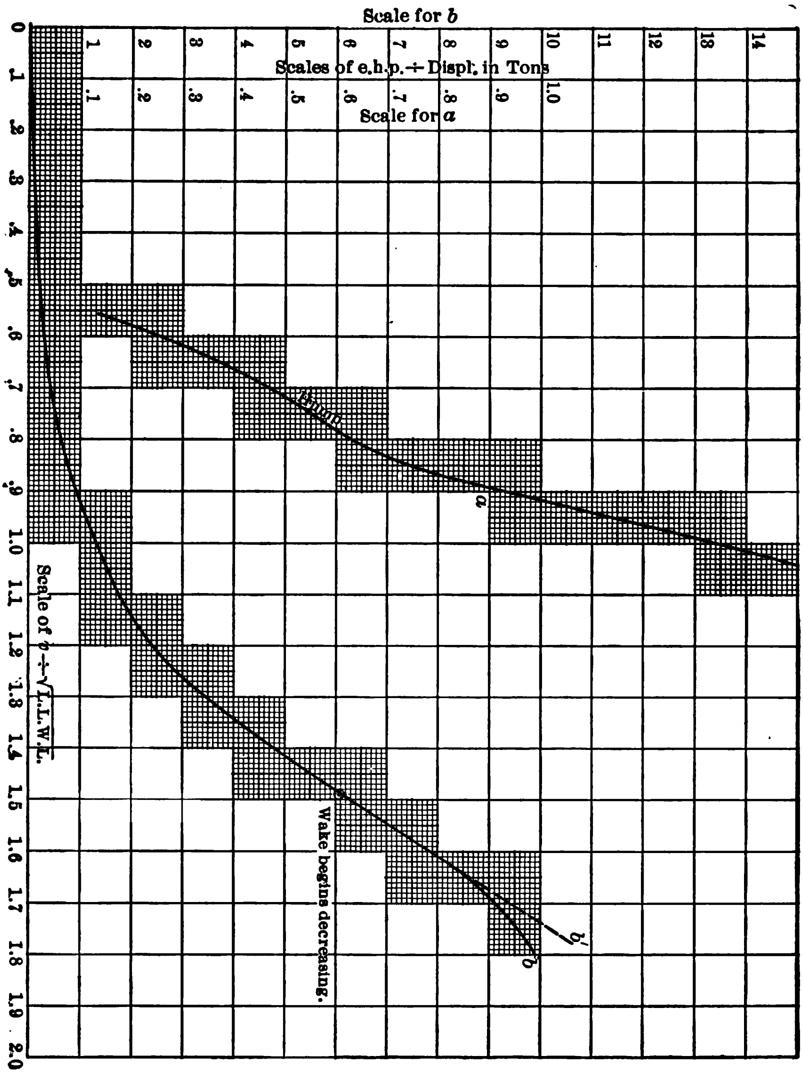


FIG. 7.—The Resistance Curve Showing Humps Caused by Abnormal Wake; also Apparent Falling Off in Resistance Caused by Abnormal Decrease in Wake.

estimating for revolutions, the speed corresponding to the power as given by the hump must be used, the actual speed for these revolutions being that as corrected by eliminating the hump.

For cases where an abnormal decrease in wake is indicated, the normal character of the curve should be extended as shown at b^1, Fig. 7, the actual speeds to be expected being those corresponding to this new curve, while the revolutions, as before, are derived from the speeds and powers given by the actual model tank curve.

When thrust deduction greater than unity exists it becomes modified by these changes in wake, being increased for the humps and decreased for the decrease. Calling the e.h.p. values of the model tank curve, e.h.p., and those for the same speed, from the corrected curve, e.h.p.$_1$, and denoting the original thrust deduction factor by K and the new one by K_1, then

$$K_1 = K \times \sqrt{\frac{\text{e.h.p.}}{\text{e.h.p.}_1}},$$

but in no case should this value of K_1 be taken as less than unity.

In many cases encountered the curves of e.h.p. will give no indication of change in wake and in such cases, unless there are performances of similar vessels at hand to use as a guide in correcting the estimate of performance as derived from the model curve, this estimate, uncorrected, must be taken and the departures from it caused by changes in wake accepted as something which can not be allowed for.

Having the actual performance of such a vessel, it becomes an easy matter to analyze and to determine whether the departure from the estimate is due to change in wake or to errors in the assumed curve of resistance, when there are no abnormalities existing in the propeller itself. Using the actual powers and speeds in the Robinson equation for apparent slip, should the apparent slips obtained correspond closely with the actual apparent slips, it is an indication that the curve of e.h.p. is too high or too low, and that the vessel is obtaining her speed without any abnormal assistance or loss from increase or decrease in wake.

Use the actual powers and the speeds corresponding to the e.h.p. values, as obtained from these actual powers and as given on the resistance curve, in the Robinson equation for apparent slips, and from these apparent slips obtain the revolutions for these resistance curve speeds. Should the resulting revolutions approximate closely to those actually obtained with these powers, the difference between the estimated performance and the actual performance will be due to wake gain or loss, depending upon whether the actual speeds obtained are greater or less than those expected from the model tank curve.

Arrangement of Strut Arms and their Influence on Wake

It has become quite the custom, at least in the United States, in the last few years to so design the strut arms that their axes lie in the direction of the lines of flow of the water around them as determined from the model tank. By comparing the performances of vessels with struts so designed, with those of similar vessels having the axes of the sections of the lower strut arms parallel to the base line of the ship, one must be lead to the conclusion that the first-named method is incorrect.

The performances of vessels having their strut arm sections placed at an angle to the base line indicate that at high speeds, such sections tend to cause the after part of the vessel to sink deeper in the water, thus broadening out the limits of the water plane. This produces an increased sternward velocity of both the high-velocity current close to the hull, also broadening this current, and of the water flowing through the propeller. This increase in velocity of flow through the propeller increases its revolutions for any given power so that the revolutions become abnormally high, and when the malign skin current becomes broad enough it enters the region of the propeller disc and entails an increase in power while, also, vibrations of more or less intensity occur.

With vessels having the strut arm sections parallel to the base line, however, the stream line currents striking the lower

sides of these arms tend to hold the after bodies up; a portion of the velocity of flow in these stream lines is destroyed due to the sudden change in the direction of flow, the revolutions are maintained more nearly normal, and where ample tip clearance is given the power also remains normal, or may even be much reduced.

It should always be borne in mind that "wake gain," so far as the action of the propeller is concerned, is equivalent to a reduction in resistance, while "thrust deduction" has exactly the same effect as an increase in resistance. To neglect "wake gain" produces an error which results in slightly overpowering a vessel for a given speed, with the result that on trial, when the designed power is developed, the speed is exceeded. This error is one for which the designer is always forgiven if the excess power is not too excessive. To seriously underestimate or to neglect the "thrust deduction," however, results in the unforgiveable offense of underpowering and the realization of a lower speed than expected. The former error always delivers a ship to a customer; the latter may throw the vessel back on the hands of her builders. Where the thrust deduction factor K exceeds 1.02, no account should be taken of the possible "wake gain."

Where "wake gain" is expected, the estimates of power and revolutions should be made by using the model tank curve of effective horse-powers and speeds, the wake gain being used only as a correction for the speed actually expected. This expected speed should not appear in either power or revolution calculations.

In the following problems, the data "Actual Power" is that actually corresponding to the estimated revolutions and "Actual Speed" is the speed which was obtained with these revolutions.

Problems in Wake Gain

Problem 15

Hull Conditions

Mean Slip B.C. for All Propellers = .632

Actual Tip Clearance = 5′

$K = 1$

Calculated values of Z are used in this and the succeeding problems.

Basic Conditions of Propellers

No. of Propellers	4
Blades	3
P.A. ÷ D.A.	.4247
D	12′.863
P	11′.209
T.S.	9000
R	223.2
$P \times R$	2502
$1 - S$	.89
V	21.97
I.T.$_D$	6.62
I.H.P. (Total)	37404
P.C.	.603
E.H.P.	22556
S.H.P.	34412

ESTIMATE OF PERFORMANCE

e.h.p. ÷ E.H.P.	.1	.2	.3	.4	.5
e.h.p.	2256	4512	6768	9024	11280
v (Tank)	11	13.86	15.75	17.25	18.4
$v \div V$	.501	.631	.717	.784	.838
Z	1.0414	.7279	.5445	.4144	.3135
K	1	1	1	1	1
S.H.P.$_d = K \times$ S.H.P.$_p$	3128	6439	9822	13253	16719
Actual Power	3250	6550	9850	13200	16750
Est. Revs.	107.8	135.9	155.2	170	182.5
Act. Revs	102.0	129.6	151.3	168	181.7
Act. Speed	11.25	14.00	16.12	17.67	18.8

e.h.p. ÷ E.H.P.	.6	.7	.8	.9	1.0
e.h.p.	13536	15788	18048	20304	22556
v (Tank)	19.3	20.05	20.7	21.15	21.45
$v \div V$	.878	.912	.942	.963	.976
Z	.231	.1613	.1009	.0477	0
K	1	1	1	1	1
S.H.P.$_d = K \times$ S.H.P.$_p$	20217	23736	27278	30833	34412
Actual Power	20300	23700	27250	31100	Off curve
Est. Revs	192	200.9	209	215	219.8
Act. Revs	190.6	198.6	206	213.5	221
Act. Speed	19.70	20.34	20.85	21.25	21.55

In the foregoing problem the "wake gain" was abnormal, and the problem is further complicated by the great variation in revolutions between the different shafts due to the differences in powers on the shafts at the various speeds. In such cases, the low-powered shafts hold down the revolutions of the higher-powered ones while the lower-powered shafts have their revolutions increased, but these increases and decreases are not in inverse proportion to each other, and this inequality produces an inequality between estimated and actual revolutions. This inequality is further exaggerated by differences between the estimated and the actual powers.

Problems in Wake Gain Showing Effect of Strut Arms

Problem 16

In this problem the vessel is nearly similar to that of Problem 17, except that there are two struts on each shaft, the axes of the lower strut arms being set at an angle of $4\frac{1}{2}°$ to the base line, inclined downward at the forward edge.

Hull Conditions

Slip B.C. = .385
Twin Screws
Large Tip Clearance
$K = 1$

As already pointed out, with struts so arranged, the stern squats badly, the squatting beginning at

$$\frac{v}{\sqrt{\text{L.L.W.L.}}} = \frac{v}{\sqrt{310}} = 1.48,$$

$$v = 26.06,$$

and being accomplished at

$$\frac{v}{\sqrt{310}} = 2.13, \; v = 37.5.$$

Basic Conditions of Propellers

Blades	3
P.A. ÷ D.A	.595
D	92″.5
P	82″
T.S.	13330
R	550.5
$P \times R$	3760
$1-S$	.787
V	29.2
I.T.$_D$	11.77
I.H.P.	18021
P.C.	.525
E.H.P	9461
S.H.P.	16580

e.h.p.÷E.H.P.	.025	.05	.075	.1	.2
e.h.p.	237	474	711	946	1892
v	10.3	12	14.3	15.52	19.25
$v \div V$	.35	.41	.49	.531	.66
Z	1.668	1.355	1.1715	1.0414	.7279
$S.H.P._d = S.H.P._p$	356	732	1117	1507	3102
Actual Power	400	720	1150	1500	3100
Est. Revs	167.2	199.2	234.6	255.9	321.1
Act. Revs	160	200	230	252	317
Act. Speed	10.3	12.55	14.3	15.52	19.45

e.h.p.÷E.H.P.	.3	.4	.5	.6	.7
e.h.p.	2838	3785	4730	5676	6623
v	21.5	22.8	24	25.1	26.15
$v \div V$	.736	.78	.822	.86	.895
Z	.5445	.4144	.3135	.231	.1613
$S.H.P._d = S.H.P._p$	4732	6385	8055	9741	11436
Actual Power	4750	6300	8100	9700	11500
Est. Revs	364.6	395.1	422.9	449.9	475.8
Act. Revs	359	394	425	453	476
Act. Speed	21.85	23.4	24.65	25.7	26.65

e.h.p.÷E.H.P.	8	.9	1.0	1.05
e.h.p.	7570	8514	9461	.9935
v	27.15	28.15	29.2	29.94
$v \div V$	.93	.964	1.0	1.02
Z	.1009	.0477	0	.0221
$S.H.P._d = S.H.P._p$	13143	14855	16580	17445
Actual Power	13100	14900	16550	17400
Est. Revs	506.7	539.5	573.2	593.2
Act. Revs	508	538	571	591
Act. Speed	27.41	28.21	29.15	29.7

Problem 17

This vessel is similar to that of Problem 16, except there is only one strut on each shaft and the axes of the sections of the lower strut arms are parallel to the base line of the vessel. Squat neglected.

Hull Condition

Slip B.C. = .385
Twin Propellers
Large Tip Clearance
$K = 1$
Propellers located at Frame 162, at after end of knuckle of keel
Vessel 12 tons lighter than model at upper speeds
Causes further increase in speed and increase in revolutions.

Basic Condition of Propellers

Blades	3
P.A. ÷ D.A.	.6013
D	7′.33
P	6′.67
T.S.	13570
R	589.3
$P \times R$	3931
$1-S$	.782
V	30.45
I.T._D	11.98
I.H.P. (Total)	16646
P.C.	.525
E.H.P.	8739
S.H.P.	15314

Act. $\frac{v}{V}$	$\frac{\text{e.h.p.}}{\text{E.H.P.}}$	e.h.p.	v (Tank)	Z	$\text{S.H.P.}_d = \text{S.H.P.}_p$	Est. Revs.	Actual Speed.	Actual. Revs.	Power.
.335	.025	219	10	1.668	329	168.6	10.2	168	350
.407	.05	437	12.0	1.355	676	203.8	12.4	203	675
.473	.075	656	14.3	1.1715	1032	243.1	14.4	237	1050
.515	.1	874	15.65	1.0414	1392	267.6	15.7	259	1400
.643	.2	1748	19.1	.7279	2866	332.6	19.6	328	2900
.716	.3	2612	21.35	.5445	4371	378.3	21.8	374	4350
.775	.4	3496	22.79	.4144	5898	413.1	23.6	413	5900
.821	.5	4370	23.98	.3135	7440	444.2	25.0	446	7400
.857	.6	5044	24.83	.231	8997	469.2	26.1	474	8900
.891	.7	6118	26.16	.1613	10563	500.0	27.15	504	10550
.951	.8	6992	27.21	.1009	12139	528.3	27.95	529	11900
.952	.9	7866	28.19	.0477	13721	559.4	29.0	564	13700
.983	1.0	8739	29.15	0.00	15314	586.1	29.9	592	15300
.99	1.05	9176	29.65	.0221	16114	597.2	30.15	607	15700
1.012	1.1	9613	30.1	.0431	16910	615.8	30.81	622	16950
1.028	1.15	10050	30.58	.0632	17713	626.1	31.3	637	17700
1.04	1.2	10487	31.03	.0825	18518	635.0	31.7	651	18500
1.054	1.25	10924	31.6	.1009	19320	651.0	32.1	668	19300

There does exist a certain amount of squat in this case but not to the same extent as in the vessel of Problem 16.

Problem 18

This vessel is similar to those of Problems 16 and 17, except there is only one strut to each shaft, the axes of the lower strut arms being inclined $6\frac{3}{4}°$ below the horizontal at the forward edge. The vessel was run on considerably higher displacement than the other two. The propellers were located about 10 ft. further aft than those of Problem 17, and slightly forward of those of 16.

Hull Conditions

Slip B.C. = .385
Twin Screws
Large Tip Clearance
$K=1$
Squatting begins at 26.06 knots and is accomplished at 37.5 knots

Basic Conditions of Propellers

Blades	3
P.A. ÷ D.A.	.6012
D	94″
P	81″.94
T.S.	13570
R	551.4
$P \times R$	3765
$1-S$	.782
V	29.06
I.T.$_D$	12
I.H.P.	19003
P.C.	.525
E.H.P	9976
S.H.P.	17483

e.h.p. ÷ E.H.P.	.025	.05	.075	.1	.2
e.h.p.	249	499	748	998	1995
v (Tank)	10	12.6	14.4	15.7	19.4
S.H.P.$_d$ = S.H.P.$_p$	376	772	1178	1589	3271
Actual Power	400	850	1300	1700	3300
Est. Revs	163.1	205.9	236.1	259.3	324.3
Act. Speed	10	12.6	14.4	15.85	20.05
Act. Revs	158	198.8	229.5	251	321

e.h.p. ÷ E.H.P.	.3	.4	.5	.6
e.h.p.	2993	3990	4988	5986
v (Tank)	21.7	23.02	24.2	25.32
S.H.P.$_d$ = S.H.P.$_p$	4990	6733	8494	10271
Actual Power	5050	6700	8500	10300
Est. Revs	367.7	398.4	425.9	454.2
Act. Speed	22.31	23.7	24.75	25.7
Act. Revs	366	398	426	454.5

e.h.p. ÷ E.H.P.	.7	.8	.9	1.0
e.h.p.	6983	7981	8978	9976
v (Tank)	26.41	27.41	28.4	29.4
S.H.P.$_d$ = S.H.P.$_p$	12059	13858	15662	17483
Actual Power	12000	13800	15605	17550
Est. Revs	482.1	513.5	544.8	577.6
Act. Speed	26.63	27.55	28.48	29.46
Act. Revs	486	518	551	583

Problem 19

This vessel was exactly similar in hull lines to those of Problems 17 and 18. It had only one strut to each shaft with the strut section axes parallel to the base line as in 17, but the struts were located in the same fore-and-aft location as those of 18. While there was undoubtedly some decrease in resistance, particularly at the high speeds, over that of 17, the effective horse-power curve of that vessel has been used in the analysis. A much better agreement between actual and estimated revolutions for equal powers will be noted in the cases of 17 and 19 than in 16 and 18, should squat be neglected, the obliquity of the strut arm axes in these two latter vessels apparently causing a decrease in pressure at the high speeds at the propeller locality causing the propellers to speed up due to the increased velocity of flow of the water (decrease in wake) through them. Where the tips of the propellers are located close to the hull in cases like 16 and 18, this undue increase in revolutions is accompanied by increasing vibration and loss in power exactly as in cases of cavitation.

Hull Condition

Slip B.C. = .385
Twin Screws
$K = 1$

Basic Condition of Propellers

Blades	3
P.A. ÷ D.A.	.611
D	87″
P	80″
T.S.	14000
R	614.7
$P \times R$	4100
$1 - S$	.779
V	31.52
I.T.$_D$	12.32
I.H.P.	18200
P.C.	.525
E.H.P.	9553
S.H.P.	16741

Actual S.H.P.$_a$=S.H.P.$_p$	1580	3080	4570	6100
Z	−1.025	−.735	−.564	−.438
e.h.p.÷E.H.P.	.102	.195	.288	.38
e.h.p.	974	1863	2751	3630
v (Tank)	16.1	19.45	21.6	23
Est. Revs.	276.4	339.3	382.4	416.6
v (Actual)	16.9	20.6	22.6	24
v÷V (Actual)	.53	.653	.717	.761
Actual Revs.	276	342	382.5	414

Actual S.H.P.$_a$=S.H.P.$_p$	7800	9570	11400	12720
Z	−.332	−.243	−.167	−.119
e.h.p.÷E.H.P.	.485	.59	.695	.77
e.h.p.	4633	5638	6639	7365
v (Tank)	24.3	25.6	26.78	27.62
Est. Revs.	450	483.7	515.3	536.5
v (Actual)	25.3	26.57	27.8	28.58
v÷V (Actual)	.803	.843	.882	.903
Actual Revs.	446	481	517.5	541.5

Actual S.H.P.$_a$=S.H.P.$_p$	14620	16680	17800
Z	−.059	−.0016	+.0266
e.h.p.÷E.H.P.	.88	.99	1.065
e.h.p.	8407	9458	10174
v (Tank)	28.8	29.92	30.71
Est. Revs.	570	601.7	621.5
v (Actual)	29.7	30.78	31.32
v÷V (Actual)	.942	.976	.993
Actual Revs.	577	611.5	628.5

The maximum difference between estimated and actual revolutions is seen to not exceed 1.6 per cent while the "wake gain" has given an increase in speed over the tank speed of from .61 to 1.02 knots.

It should be borne in mind that these vessels being of the destroyer type, their resistance is affected very materially by changes in load and trim, and that in trying them over the trial course the loads are usually considerably heavier at the beginning of the trials than at the end. This variation in load also has its effect on revolutions.

Problem 20. Propellers for "Tunnel" Boats

By the term "Tunnel" Boat is meant a vessel of shallow draft having arched passages formed in the bottom of the after body and the propellers located in these tunnels. The propeller may be so located that only a portion of its diameter is immersed when at rest. When in motion the propeller draws the air from the portion of the tunnel forward of it and expels it to the rear. This produces a vacuum which is immediately filled by water, the tunnel thus remaining full so long as the propeller is operating.

The water being constrained to move in a direction practically normal to the disc of the propeller, the principal losses are those due to friction in the tunnel and are therefore practically a constant percentage loss of the total power put into the propeller, no matter what the slip block coefficient of the ship may be.

This loss is heavy, and from the results obtained in the following problem, appears as "thrust deduction" and amounts to $K = 1.195$. The problem is an analysis of the propellers of two U. S. shallow water gunboats for which the propellers were designed by parties who have had much experience with this type of vessel and whose design of propeller must, therefore, be considered as having been based upon actual performances.

The close agreement between the analysis results and the designed conditions is rather good evidence as to the correctness of the former and of the value of K obtained.

Nom. B.C. = .6		800 (Two engines) = I.H.P.$_a$
$B \div$ L.W.L. = .153	PA ÷ DA	Revs. 300
Slip B.C. = .575	"	$v = 13\frac{1}{4}$ knots
$\frac{4}{3}$ P.A. ÷ D.A. 347	.26025	
D . . . . 4′.67		
P . . . . 6′		
T.S. 5730		
R 390.6		
$P \times R$ 2343		
$1 - S$ 90		
V 20.81		
I.T$_D$ 2.93		
I.H.P. 1187		
P.C. 655		
E.H.P. 777		
v 13.25		
e.h.p. 385		
e.h.p. ÷ E.H.P. 4954		
$v \div V$ 637		
$(E.T. \div \text{e.t.})^x$ . . . . .86 (Sheet 22)		
Z 314		
i.h.p.$_p$ 576		
i.h.p.$_p$ ÷ $(E.T. \div \text{e.t})^x$. 669.8		
$K = 800 \div 669.8 = 1.195$		

The designed conditions of these vessels were

$$\text{I.H.P.}_d = 800,\ \text{Revs.} = 300,\ \text{Speed} = 13\tfrac{1}{2}\ \text{knots.}$$

The apparent slip with these conditions = 25.41 per cent.

The computed apparent slip, from the basic condition reduced to $13\frac{1}{2}$ knots and 800 I.H.P., is as follows:

$$\begin{aligned} \log A_V &= 3.93 \\ S &= .10 \\ \text{I.H.P.}_d &= 800 \\ \log A_v &= 3.36 \\ s &= \text{Apparent slip} = .2504 \\ \text{Revs.} &= 298.5 \end{aligned}$$

Problem 21. Double-ended Ferry Boat—Propellers

In vessels of this type the form of hull is such that the midship section coefficient is usually much finer than standard for the slip-block coefficient. No correction of slip block coefficient should, however, be made for this.

Of the two propellers, the after one is that which works in standard propeller conditions and the analysis, therefore, applies to this propeller. The difference between the actual horse-power of the engine and that derived by the analysis for the after propeller is credited to the forward one.

It will be noted that the analysis indicates that the after propeller delivers 63.66 per cent of the total effective horse-power and absorbs in doing this about 55 per cent of the total power of the engine, while the forward propeller only delivers 36.34 per cent of the effective power at an expense of 45 per cent of the engine power. This inefficiency of the forward screw would lead to the belief that the efficiency of propulsion would be greatly increased if the forward propeller were uncoupled and allowed to revolve freely, or, better still, if it were removed completely.

This expected betterment has been realized by actual experiment and the analysis of such screws in the following problem also promises such a result, a reduction in total indicated power from 1845 I.H.P. to 1570 I.H.P. being shown, a reduction in the required power of 14.9 per cent.

Double-ended Ferry Boat

Hull Conditions	*Basic Conditions of Propellers*	
Slip B.C. = .76	Blades	4
Two propellers—one forward, one aft	$\frac{4}{3}$ P.A. ÷ D.A.	.432
K = 1.43	P.A. ÷ D.A.	.324
Draft = 13 feet	D	10′.5
	P	9′.55
	T.S. for P.A. ÷ D.A.	7150
	R	216.8
	$P \times R$	2070
	$1 - S$ for P.A. ÷ D.A. = .324 and slip B.C. = .76 =	.93
	V	19.00
	I.T.$_D$ for P.A. ÷ D.A.	4.25
	I.H.P. = $(D^2 \times \text{I.T.}_D \times P \times R) \div 252.41$	3843
	P.C. for $\frac{4}{3}$ (P.A. ÷ D.A.)	.598
	E.H.P	2298

Analysis

V	19.00
log A_V	3.82
v	14.662 miles
v	12.73 knots
log A_v (Curve Y, sheet 21.)	3.15
Actual Revs	147.9
s = Apparent slip	.0865
$\text{I.H.P.}_a = K \times \text{I.H.P.}_p = s\dfrac{\text{I.H.P.} \times v^y}{S \times V^y}$*	1015
Total I.H.P.$_a$ = Total $K \times$ I.H.P.$_p$	1845 = Actual total power
Power on forward screw	830 = 1845 − 1015
K	1.43
I.H.P.$_p$	709.8
e.h.p.	700 (Total by two screws)
Z	.73342
e.h.p. aft. ÷ E.H.P	.194
e.h.p. aft.	445.6
e.h.p. fd.	254.4 = 700 − 445.6
$v \div V$	.67

*$v^y = A_v$ and $V^y = A_V$.

Forward Propeller Removed

e.h.p.	700
e.h.p. ÷ E.H.P	.3046
Z	.5440
$\text{I.H.P.}_a = K \times \text{I.H.P.}_p$	1570
$v \div V$	.67

and the propeller plots well within the safe zone on Sheet 22 and for safe loads on Sheet 22B.

Problem 22

Use of Sheet 22 in estimating power and effective power delivered.

By the aid of the curves given on this sheet it becomes possible to make an estimate of the indicated or shaft horse-power being developed by the engines and of the effective horse-power being delivered by the propeller, provided the characteristics of the hull and of the propeller together with the revolutions necessary for any speed are known, thus:

Suppose the vessel given in Column 1, page 81, be taken:

The slip block coefficient of the hull is .805.

Thrust deduction factor K is 1.27.

Basic apparent slip is .055.

By analysis of the propeller, the Basic I.H.P. is 7996 and the Basic E.H.P. is 5061, while the Basic $V = 14.43$ knots.

Suppose the ship be so loaded down that on account of bad weather and head wind and sea a speed of 10 knots is made with 80 revolutions. The pitch of the propeller being 15 ft., the apparent slip will be

$$\frac{(15\times 80)-(10\times 101.33)}{15\times 80} = .156 = s.$$

From Sheet 21, log A_V for 14.43 knots is 3.47, while log A_v Curve X, for 10 knots is 3.00.

The Robinson equation for apparent slip in terms of power is

$$s = S\times\frac{\text{I.H.P.}_a\times A_V}{\text{I.H.P.}\times A_v},$$

therefore,

$$\text{I.H.P.}_a = s\times\frac{\text{I.H.P.}\times A_v}{S\times A_V} = .156\times\frac{7996\times 3.0}{.055\times 3.47} = 7864.$$

Now the value of $v \div V = 10 \div 14.43 = .693$.

The curve of critical thrusts, $E.T.$, cross the line of $v \div V = .693$ at e.h.p. ÷E.H.P. = .610, and this value is taken as the starting point for estimate of e.h.p. ÷ E.H.P. being delivered by the propeller.

Following along the line of .693 and taking the points where this line crosses each curve of $(E.T. \div \text{e.t.})^2$, the following values of e.h.p. ÷ E.H.P. and $(E.T. \div \text{e.t.})^2$ are obtained:

$$\text{e.h.p.} \div \text{E.H.P.} = \begin{cases} .610 \\ .67 \\ .72 \\ .80 \end{cases} \text{and } (E.T. \div \text{e.t.})^2 = \begin{cases} 1.0 \\ .95 \\ .90 \\ .85 \end{cases}$$

Turning again to the Robinson equation, but using the second form, namely,

$$s = S\frac{K \times Av}{10 \times A_0} \times \left(\frac{\text{e.t.}}{E.T.}\right)^2,$$

and using the above values of $(E.T. \div \text{e.t.})^2$, a series of apparent slips is obtained as follows:

$$s \text{ for } \left\{ \begin{matrix} \text{e.h.p.} \div \text{E.H.P.} = .610 \\ \text{e.h.p.} \div \text{E.H.P.} = .67 \\ \text{e.h.p.} \div \text{E.H.P.} = .72 \\ \text{e.h.p.} \div \text{E.H.P.} = .80 \end{matrix} \right\} = \left\{ \begin{matrix} .1225 \\ .1434 \\ .1622 \\ .1922 \end{matrix} \right\} \text{ where } Z = \left\{ \begin{matrix} .226 \\ .18 \\ .15 \\ .1009 \end{matrix} \right.$$

Laying down these values of apparent slips as curves having as ordinates values of apparent slip and as abscissas values of e.h.p. ÷ E.H.P., the value of e.h.p. ÷ E.H.P. corresponding to an apparent slip of .156 per cent is found to be .7035.

Therefore the effective horse-power being delivered equals

$$\text{e.h.p.} = \text{E.H.P.} \times \frac{\text{e.h.p.}}{\text{E.H.P.}} = 5061 \times .7035 = 3560$$ and the propulsive coefficient realized equals

$$\text{p.c.} = \text{e.h.p.} \div \text{I.H.P.}_a = 3560 \div 7864 = .452+.$$

The only cause of any appreciable error occurring, is the value of K which must always remain a cause as it is dependent upon the form of hull and location of the propeller. Errors in the value of this factor affect the value of e.h.p. obtained and, therefore, the value of the propulsive coefficient realized.

Should the propeller have fallen on or above the curve of critical thrusts, *E.T.*, the log I.H.P.$_{a}$ would have equalled

$$\log \text{I.H.P.}_a = \log \text{I.H.P.} - Z\left(\text{for } \frac{\text{e.h.p.}}{\text{E.H.P.}} = .7035 \text{ is } .16\right) + \log K$$

$= 3.90287 - .16 + .10380 = 7025$ and the propulsive coefficient

$=$ p.c. $= 3560 \div 7025 = .506+$ and this would have been realized where

$v \div V \overline{\gtrless} 710$ which corresponds to a speed of not less than

$$14.43 \times .710 = 10.25 \text{ knots.}$$

Problem 23. Analysis of Performance of Submarine Boat Propellers

In the following work the performances of five separate vessels are given, three of them being of the single hull, Holland type, with the propellers carried abaft and clear of the hull while the remaining two are of the double hull, Lake type, the propellers being carried below and in close proximity the hull.

HULL CHARACTERISTICS

Propeller.	L.L.W.L.	Vessel.		$B \div$ L.W.L.	Nom. B.C. Surf. Cond.	Slip B.C. Surf.	B.C. Subm.
		Beam $= B$	Draft $= H$				
A	153'.5	16'.167	13'.5	.1053	.444	.745	.745
B	167'.42	17'	13'.583	.1016	.4083	.73	.73
C	147	15'.25	12'.5	.1038	.4327	.737	.737
D_1, D_2, D_2, D_3, D_4	165'	14'.75	13'.25	.0894	.4784	.585	.80
E	155'	14'.0	12'.33	.09032	.4327	.575	.79

The propellers used were as follows:

A—Oval blades, broader at tip than standard;
B—Same propeller as A;
C—Standard form of blades;
D_1—Standard form of blades;
D_2—D_1 with three inches cut off diameter;
D_3—Standard form of blades. Blades of cast iron, rough and untrue;
D_4—Standard form of blades;
E—Standard form of blades.

All the above propellers were of bronze with the exception of D_3, were highly polished and sharpened at the edges, and were 3-bladed.

BASIC CONDITIONS OF PROPELLERS

Prop.		A	B	C	D_1	D_2	D_3	D_4	E	
Diam.	Act.	55"	55"	53"	63"	60"	60"	60"	51"¾	
	Basic	63"	63"	53"	63"	63"	60"	60"	51"¾	
P		45"	45"	48".5	61".5	61".5	54"	59"	45"	
P.A. / D.A.	Act.	.363	.363	.297	.268	.274	.35	.35	.4	
	Basic	.5	.5	.297	.268	.268	.35	.35	.4	
T.S.		10550	10550	6600	5950	5950	7660	7660	8600	
$P \times R$		2399	2399	1923	1849	1849	2195	2296	2399	
Condition		Surf.	Subm.	Subm.	Subm.	Subm.	Subm.	Subm.	Subm.	Surf.
S.B.C.		.745	.73	.737	.80	.80	.80	.80	.79	.575
$1-S$		.899	.8973	.926	.945	.945	.939	.939	.928	.882
V		21.28	21.24	17.59	17.24	17.24	20.34	21.28	21.97	20.66
I.T.D		8.75	8.75	3.7	.323	3.23	4.83	4.83	6.05	
I.H.P		1983*	1983*	475.8	564.1	511.7*	908.1	950.1	910.8	
P.C.		554	.554	.683	.695	.695	.652	.652	.618	
E.H.P		.837*	837*	324.8	392	355.6*	592.1	619.5	563	
S.H.P		1390*	1390*	437.5	519	470.7*	835.4	874.1	838.1	

* These quantities have been corrected for the reduction in diameter by multiplying the basic values by $\left(\frac{\text{Actual diameter}}{\text{Basic diameter}}\right)^2$.

In estimating for revolutions in these cases take the values log A_V in all cases from the curve X, Sheet 21, while for the Holland boats on the surface log A_e is taken from the X curve, it is taken from the Y curve for the submerged condition and is taken from the same corresponding curve for both surface and submerged conditions for the Lake boats. Where Holland type is loaded down by the stern or of such form of hull as to produce a heavy squat, the values of log A_e shift from X to Y exactly as occurs in the case of destroyers, which squat. At what speed the values of log A_e begin to pass from X to Y (Sheet 21), depends entirely upon the running actual trim and at the present writing, due to lack of data the approximate speed ratio due to squat cannot be given.

With the Holland type of ves el the wake appears to be variable in the surface condition, rapidly increasing with the speed so that the value of log A_e is more nearly correct for speeds below 9 knots when taken from the Y curve, but shifts gradually to the X curve in passing from about 8.5 to 10 knots, after which it practically remains constant.

The estimates of performance and the comparison with the actual performance are given in the following tables.

v	e.h.p.	$\frac{\text{e.h.p.}}{\text{E.H.P.}}$	Z	$v+V$	$\left(\frac{E.T.}{\text{e.t.}}\right)^z$	K	S.H.P.$_p$	S.H.P.$_d$ Est.	S.H.P.$_d$ Act.	S	Log AV	Log Av_1	s	Revs. Est.	Revs. Act.
							Vessel A. Surface								
8.5	50	.0597	1.29	.3994	1	1	71.5	71.5	76	.101	3.96	2.85	.06693	246.2	255
9.0	61	.0729	1.18	.4229	1	1	92	92	96	.101	3.96	2.91	.0750	262.9	270.5
10.0	81	.0968	1.06	.4698	1	1	121	121	120	.101	3.96	3.02	.0766	292.6	298
11.0	103	.123	.95	.5168	1	1	156	156	150	.101	3.96	3.13	.0766	321.9	323.5
12.0	126	.150	.855	.5638	1	1	194	194	184	.101	3.96	3.23	.0757	350.8	350
12.5	142	.170	.8	.5873	1	1	220	220	210	.101	3.96	3.29	.0748	365.1	364.5
							Vessel B. Submerged								
5	20	.024	1.66	.235	1	1	30.4	30.4	30	.103	3.96	1.92	.2438	178.7	179
6	34	.041	1.4	.282	1	1	55.3	55.3	57.5	.103	3.96	2.19	.2509	216.4	213.5
7	53	.063	1.25	.33	.900	1	78.2	86.9	92	.103	3.96	2.4	.233	246.6	248
8	76	.091	1.04	.376	.90	1	107.2	140.9	133	.103	3.96	2.57	.2555	290.4	283
9	106	.127	.94	.413	.850	1	159.6	187.8	188	.103	3.96	2.72	.2411	320.5	321
10	143	.171	.79	.47	.91	1	225.4	247.7	248	.103	3.96	2.85	.235.8	353.6	355
10.5	164	.196	.74	.5	.96	1	253	263.5	271	.103	3.96	2.92	.2135	360.8	369.5
							Vessel C. Submerged								
5															
6	29	.089	1.09	.3414	.70	1	35.56	50.8	50	.074	3.72	2.19	.2912	212.2	211.8
7	45	.137	.9	.3983	.68	1	55.08	81.0	81	.074	3.72	2.4	.2863	245.9	249
8	66	.203	.715	.4552	.68	1	84.32	124.0	120	.074	3.72	2.57	.2963	285.0	283.5
9	94	.287	.562	.5122	.70	1	119.9	171.3	176	.074	3.72	2.72	.2897	317.7	316
10	126	.388	.43	.569	.70	1	162.6	232.3	233	.074	3.72	2.85	.2846	350.5	348
10.5	145	.461	.357	.5975	.73	1	192.3	263.4	262	.074	3.72	2.92	.2811	366.2	364.5

VESSEL D_1. SUBMERGED S.B.C. = .80. (Tips close to hull and to each other)															
8	87	.2219	.679	.4639	.675	1.17	109	188.4	185	.07	3.70	2.57	.2693	216.5	220
9.6	157	.4005	.4144	.5567	.67	1.17	200	349.1	360	.07	3.70	2.80	.3222	280.1	287
VESSEL D_2. SUBMERGED S.B.C. = .80															
8	85.26	.2398	.642	.4639	.645	1.17	107	194.7	187	.07	3.70	2.57	.3069	228.2	239
9.42	148	.4162	.4	.5463	.645	1.17	187	339.9	357	.07	3.70	2.76	.3459	284.8	282
VESSEL D_3. SUBMERGED S.B.C. = .80															
*8	87	.145	.87	.3933	.65	1.0	112.7	173.4	182	.08	3.91	2.57	.277	249.2	267
9.77	170	.2871	.57	.4803	.62	1.0	225	362.7	375	.08	3.91	2.81	.3412	334	325
VESSEL D_4. SUBMERGED S.B.C. = .80															
8	85.26	.1376	.90	.376	.62	1.0	110	177.5	155	.08	3.96	2.57	.3041	247.4	245
10.18	170	.2744	.583	.4783	.625	1.0	227	363.1	368	.08	3.96	2.87	.3160	320.3	322
VESSEL E. SUBMERGED S.B.C. = .79															
													†.3974	358.7	
8	75	.1332	.91	.3642	.60	1.0	171.9	171.9	155	.072	4.0	2.57	*.3564	336.9	340
9.29	109	.1936	.74	.4229	.61	1.0	250	250	251	.072	4.0	2.76	.3733	400.6	395
VESSEL E. SURFACE S.B.C. = .575. (Squatting)															
8	47	.08349	1.12	.3642	1	1.0	63.57	63.57	56	.072	4.0	2.57	.1845	262.7	263
10.5	95	.1688	.8	.478	1	1.0	132.8	132.8	127	.072	4.0	2.92	.1875	349.2	350

* By using actual power. † By using estimated power.

CHAPTER VIII

CAVITATION

TAYLOR, in his work on "Speed and Power of Ships," states as follows concerning this phenomenon:

"*Nature of Cavitation.* The phenomenon known as cavitation has been given much attention, of late years, in connection with quick-running turbine-driven propellers. It appears to have been first identified upon the trials, in 1894, of the torpedo boat destroyer *Daring*, which had reciprocating engines. When driven at full power with the original screws this vessel showed very serious vibration, evidently due to some irregular screw action. The propulsive efficiency was poor, the maximum speed obtained being 24 knots for 3700 I.H.P., and 384 revolutions per minute.

"Mr. Sidney W. Barnaby, the engineer of the Thornycrofts, who built the *Daring*, came to the conclusion that at high thrust per square inch at which the screws were working the water was unable to follow up the screw blades, and that 'the bad performance of the screws was due to the formation of the cavities in the water forward of the screw, which cavities would probably be filled with air and water vapor.' So Mr. Barnaby gave the phenomenon the name of cavitation. The screws which gave the poor results had a diameter of 6 ft. 2 in., a pitch of 8 ft. $7\frac{3}{4}$ in., and a blade area of 8.9 sq. ft. Various alternative screws were tried, and the trouble was cured by the use of screws of 6 ft. 2 in. in diameter, 8 ft. 11 in. pitch, and 12.9 sq. ft. blade area. With these screws 24 knots was obtained with 3050 I.H.P., and the maximum speed rose from 24 knots to over 29 knots."

For the *Daring* cavitation appeared to begin when the screw area was such that the thrust per square inch of projected area was a little over 11 lb. per square inch; $11\frac{1}{4}$ lb. is the figure given by Mr. Barnaby. "For a time it was thought that the thrust per square inch of projected area was a satisfactory criterion in connec-

tion with cavitation, and that the limiting thrust per square inch of projected area found on the *Daring* was generally applicable.

"This, however, is not the case. Greater thrusts have been successfully used and cavitation is liable to appear at much lower thrusts. In one case, within the author's experience, cavitation appeared when the thrust was about 5 lb. per square inch of projected area, the tip speed being about 5000 ft. per minute, and in another when the thrust was about 7.5 lb. and the tip speed about 6500 ft. per minute. There is little doubt that the prime factors involved in cavitation are: (1) The speed of the blade through the water, which is evidently measured by the tip speed, and (2) the shape of the blade section."

While Chief Constructor Taylor may be correct in his statements as to the prime factors involved in cavitation, it would appear that as all of the elements of the propeller, namely, pitch, diameter, projected area and revolutions, and in addition, the form of the afterbody of the vessel behind which the propeller is working, each has its influence, it would be difficult, if not impossible, to differentiate between them as to their relative effect. Also, it is considered that the prime factor in regulating the thrusts that can safely be used on any particular propeller is the form of the afterbody of the hull. As the afterbody fines, the thrust may be increased, and vice versa. Thus Sheet 20 is derived from the actual performance of numerous vessels, and the values of $E.T._p$ derived from it for the different standard block coefficients are considered as those which can safely be used without noticeable cavitation occurring when no thrust deduction exists. It is safe to exceed these Sheet 20 thrusts by 20 per cent without the vibration from cavitation becoming excessive.

The thrusts given on Sheet 20 and on Sheet 22 are as unaffected by thrust deduction.

Where the speed is less than the Basic speed and the entire conditions of resistance changed from the Basic conditions, the line of equal condition of effective thrusts with those of the Basic conditions is shown on Sheet 22. An overthrust of 20 per cent on these conditions may be allowed with safety, as although this overload will surely put the propeller in the cavi-

tating zone, the vibrations will not be serious and the loss of power will be slight.

Should the propeller be working in conditions where "thrust deduction" exists, however, the value of $v \div V$ at which cavitation will occur will be much higher than where no "thrust deduction" exists.

The power necessary to give a certain net e.h.p. without "thrust deduction" has been shown to be H.P.$_{p}$. Where "thrust deduction" exists, this power becomes K.H.P.$_{p}$=H.P.$_{d}$.

If no "thrust deduction" existed the power K.H.P.$_{p}$ would deliver a gross effective horse-power of e.h.p.$_1$, the value, log H.P. (Basic) − log K.H.P.$_{p}=Z$, being considerably less than log H.P. (Basic) − log H.P.$_{p}$ and therefore e.h.p.$_1$, would be considerably greater than e.h.p.

Now this greater power is spent on the water passing through the propeller and requires a higher number of revolutions (R_d), to absorb it. This increased number of revolutions demands an increased flow of water to the propeller over that required for any speed v obtained with H.P.$_{p}$ and R_p revolutions. If this increase in flow is not provided cavitation occurs. That is, where "thrust deduction" exists cavitation occurs at much lower speeds, and nominally lower effective thrusts than where no "thrust deduction" exists.

Cavitation does not depend upon tip speed because if a vessel is running under certain load conditions at a certain speed without cavitation, if she be loaded down sufficiently to produce a considerable diminution of speed for the same engine power, cavitation may ensue and yet the revolutions of the propeller and consequently the tip speed may be considerably lower than in the original condition. Should increase in power be met by a corresponding increase in speed, in other words should the apparent slip not rise abnormally for increase in power, cavitation, in the opinion of the writer, would never occur so long as the effective thrusts were held down. Information has been received very lately of a vessel steaming at a speed of over 39 knots, the tip-speeds of the propellers exceeding 17,000 ft. per minute, with no evidences of cavitation existing.

As to the influence of blade section on cavitation, should the section of the blade be so bad as to prevent the water engaging and leaving the blade freely, false cavitation may be produced by excessive eddying in the blade-section wake. Further, should the section be of normal form but abnormally thick in comparison with the blade width, the actual pitch will be increased very considerably above the nominal pitch, the basic V will be increased and the speed factor $v \div V$ for any speed v and load factor e.h.p. ÷ E.H.P., will become smaller, thus bringing the propeller nearer the cavitating point for any value of e.h.p. ÷ E.H.P.

There is also a phenomenon encountered in the cases of vessels subjected to great variation of resistance with practically constant power in the engines which is analogous in its effects to cavitation. This condition is illustrated on Sheet 22.

On this sheet are shown two curves marked "Lower Limit of e.t." and "Curve of Critical Thrusts, *E.T.*" This latter curve will be called the curve of *critical thrusts*.

Propellers as designed for any particular resistance of ship should usually fall between these two limiting curves, and the curve of performance of the vessel at the hull condition corresponding to the resistance used in the design would then fall between these two limits.

Now, suppose the vessel to be a tow-boat or a slow-speed, low-powered merchant ship, both of which are subject to great variations in loading.

Suppose that the propeller be designed to deliver the necessary e.h.p. for 10 knots, at a load factor of e.h.p. ÷ E.H.P. = .3 and at a speed factor $v \div V = .74$ (See Sheet 22). This point will then fall on the curve marked "Lower Limit of e.t." The corresponding value of V will be $10 \div .74 = 13.65$.

Assuming that the value of $K = 1.3$, and of the basic slip 6 per cent, the I.H.P. necessary for the speed will be

$$\text{I.H.P.}_{a} = K \times \text{I.H.P.}_{v} = \text{I.H.P. (Basic)} \div 10^{z} \times K$$
$$= \text{I.H.P.(Basic)} \div 10^{.5445} \times 1.3,$$

and the apparent slip =

$$s = S \times \frac{\text{I.H.P.}_{a} \times V^{y}}{\text{I.H.P.} \times v^{y}} = .06 \times \frac{\text{I.H.P.}_{a} \times \overline{13.65}^{y}}{\text{I.H.P.} \times 10^{y}}.$$

Now, suppose the vessel to be so loaded down or the tow boat to take such a tow that with the same e.h.p. a speed of only 7.5 knots can be realized. As the load is gradually increased and the speed decreases with this increase, the propulsive efficiency of the propeller remains practically constant while the apparent slip increases slowly until the limit marked "Curve of Critical Thrusts, *E.T.*" is reached. As the load is still further increased and the speed factor $v \div V$ falls below the value on e.h.p. $\div$ E.H.P. $=.3$, corresponding to *E.T.*, a new factor enters into the power and slip equations due to a dispersal of the thrust column flowing from the propeller being produced. This dispersal of the thrust column makes necessary an augmented flow to and through the propeller and this increased demand for supply carries with it an augment of power and of revolutions. The inverse of these augments are shown as curves on Sheet 22, marked "Curves of $\left(\frac{E.T.}{\text{e.t.}}\right)^{x}$."

The new value of the power required to deliver the original e.h.p. becomes

$$\text{I.H.P.}_{a}=K\times \text{I.H.P.}_{v}\times\left(\frac{\text{e.t.}}{E.T.}\right)^{x}=\text{I.H.P.}\times K\times\left(\frac{\text{e.t.}}{E.T.}\right)^{x}\div 10^{z},$$

or

$$\text{I.H.P.}_{a}=1.3\times \text{I.H.P.}\times\left(\frac{\text{e.t.}}{E.T.}\right)^{x}\div 10^{.5445}.$$

When v has dropped to 7.5 knots, $v \div V=.549$ and the corresponding point on the e.h.p. $\div$ E.H.P. $=.3$ ordinate falls on the curve of $\left(\frac{E.T.}{\text{e.t.}}\right)^{x}=.80$,

Therefore,

$$\text{I.H.P.}_{a}=\frac{1.3}{.8}\times \text{I.H.P.}\div 10^{z},$$

and the equation for apparent slip becomes

$$s=.06\frac{\text{I.H.P.}_{a}\times V^{y}}{\text{I.H.P.}\times v^{y}}=s\frac{1.3\times \text{I.H.P.}\times \overline{13.65}^{y}}{.8\times \text{I.H.P.}\times \overline{7.5}^{y}\times 10^{.5445}}.$$

This new value $K_1=K\times\left(\frac{\text{e.t.}}{E.T.}\right)^{x}=\frac{1.3}{.8}$ may be called an augmentation of the Basic thrust deduction.

The new value of the propulsive coefficient, which was originally

$$\text{p.c.} = \frac{\text{e.h.p.}}{\text{K} \times \text{I.H.P.}_p},$$

has now become

$$\text{p.c.} = \frac{\text{e.h.p.}}{K_1 \times \text{I.H.P.}_p} = \frac{\text{e.h.p.}}{K \times \left(\frac{\text{e.t.}}{E.T.}\right)^2 \times \text{I.H.P.}_p}.$$

Cavitation depends upon after body, projected area ratio, effective thrust and thrust deduction and no other conclusion can be arrived at from the evidence at hand.

In support of this contention the cases of three identical vessels, identical as to hulls but fitted with different propellers, is here given: The effective horse-power curve has been derived from the performance of that vessel where there is absolutely no doubt that cavitation did not exist, and the performances of the other two vessels were then estimated from this derived curve, although the third one of the vessels ran at 10 tons heavier displacement than the other two.

Problem 24

Ship	*Paul Jones*	*Perry*	*Preble*
Propellers	2	2	2
Blades	3	3	3
P.A.÷D.A	.43	.275	.358
D	7′.42	7′.42	7′.42
P	10′.67	10′.833	10′.42
T.S.	9110	6100	7800
R	390.8	261.7	334.6
P×*R*	4170	2835	3487
Slip B.C.	.31	.31	.31
1−*S*	.829	.845	.838
V	34.04	23.64	28.84
I.T.$_D$	.678	3.22	5.01
I.H.P.	10668	3445	6592
P.C.	.60	.694	.647
E.H.P.	6401	2391	4265

The estimates of e.h.p. and of performances, basing these upon the values of e.h.p. derived from the performance of the *Paul Jones*, will now be made.

PAUL JONES

$I.H.P._d = I.H.P._p$	Z	$\frac{e.h.p.}{E.H.P.}$	e.h.p.	v
700	1.183	.073	467	16
1200	.949	.123	787	18
2000	.727	.201	1287	20
3150	.530	.309	1978	22
4600	.365	.45	2881	24
5300	.304	.515	3297	25
6150	.239	.584	3738	26
6800	.196	.655	4193	27
7350	.162	.699	4474	28
7600	.147	.722	4622	28.5
$K=1$				

PERRY

$\frac{e.h.p.}{E.H.P.}$	Z	$I.H.P._d = I.H.P._p$		$v \div V$	
		Est.	Actual.	Actual.	Cavit. e.t. = 1.15 E.T.
.1954	−.734	636	600	.677	.1699
.3293	−.500	1089	995	.761	.2863
.5381	−.284	1791	1700	.846	.4679
.8272	−.088	2813	2750	.931	.7183
1.205	+.083	4171	4500	1.015	1.048
1.379	+.145	4811	5500	1.058	1.199
1.564	+.200	5460	6500	1.100	1.36
1.754	+.254	6183	7600	1.142	1.525
1.871	+.283	6610	8750	1.185	1.627
1.933	+.2981	6844	9300	1.206	1.681

PREBLE

$\frac{\text{e.h.p.}}{\text{E.H.P.}}$	z	$\text{I.H.P.}_d = \text{I.H.P.}_p$		$v + V$	
		Est.	Actual.	Actual.	Cavit. e.t. = 1.15 E.T.
.1095	−1.000	659	1000	.555	.0952
.1846	−.759	1148	1500	.624	.1605
.3015	−.540	1901	2250	.694	.2622
.4638	−.353	2924	3400	.763	.4033
.6754	−.180	4355	4800	.832	.5873
.7729	−.116	5047	5500	.867	.6721
.8765	−.06	5742	6150	.902	.7622
.983	−.004	6532	6750	.936	.8548
1.049	+.022	6935	7200	.971	.9123
1.084	+.038	7195	7400	.988	.9423

The *Preble* was run at a heavier displacement, with slightly rougher bottom, and in a little worse weather conditions than the *Paul Jones* and *Perry*, and these differences of conditions account for the differences between estimated and actual powers for that vessel.

Turning to Fig. 8, a curve of percentage increase in power for the *Perry* is shown due to the effect of cavitation. This curve is based on values of $\frac{\text{e.t.}}{\text{E.T.}}$ as abscissas, e.t. being the actual values of effective thrust, while E.T. are the Basic design conditions of this thrust. The estimate of power given in the table is without the effect of cavitation taken into account.

In order to estimate accurately the factor of increase to use for cavitation, and also as a guide to prevent entering the cavitation range, Sheet 22 has been prepared. This sheet has as ordi-

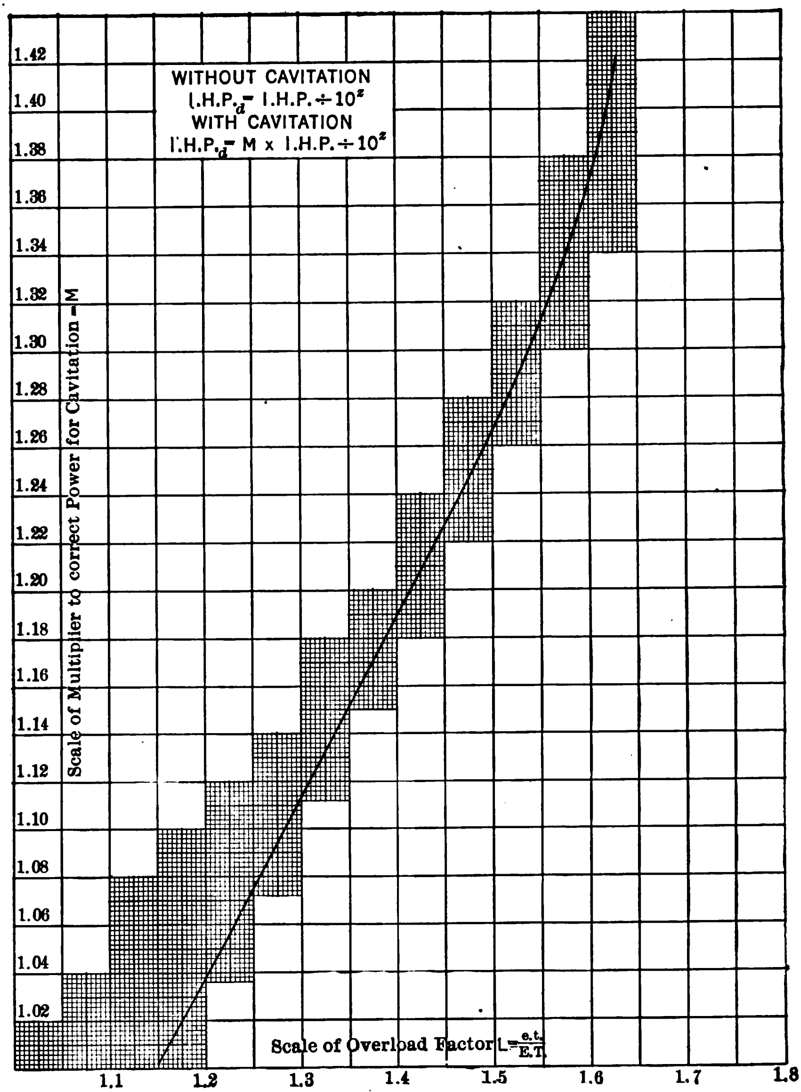


FIG. 8.—Curve of M for Augment of Power Due to Cavitation.

nates values of $\frac{v}{V}$, while the abscissas are values of $\frac{\text{e.h.p.}}{\text{E.H.P.}}$ The diagonal lines show the varying values of $\frac{v}{V}$ and of $\frac{\text{e.h.p.}}{\text{E.H.P.}}$ for values of $\frac{\text{e.t.}}{\text{E.T.}}$ from 1 to 1.75. The line of $\frac{\text{e.t.}}{\text{E.T.}}=1$, is that where the actual effective thrusts are equal to the Basic effective thrusts of the design condition. Cavitation of the suction column, however, does not begin until E.T. equals approximately 1.15.

When this condition of thrust is reached, the actual values of Z, instead of following the mathematical curve of Z, Sheet 21, pass off from it approximately on the tangent to the curve at this point.

The equation to the tangent is

Tan $\theta = -\dfrac{1.0414}{\left(\dfrac{\text{e.h.p.}}{\text{E.H.P.}}\right)}$ where θ is the angle made by the tangent with the axis of abscissas.

The new values of Z, which denote as Z_1, may also be calculated as follows:

Calling M the power correcter as ascertained from Fig. 8, for the value of $\frac{\text{e.t.}}{\text{E.T.}}$, the equation for power becomes

$$\text{I.H.P.}_{v} = M \times \text{I.H.P.} \div 10^{z} \text{ or}$$

$$\text{Log I.H.P.}_{v} = \log \text{I.H.P.} + \log M \pm Z,$$

therefore $Z_1 = \log M \pm Z$, Z being additive when $\frac{\text{e.h.p.}}{\text{E.H.P.}}$ is greater than unity.

The values of $\frac{\text{e.h.p.}}{\text{E.H.P.}}$, corresponding to these values of Z_1, should be used as abscissa values of Sheet 22, in ascertaining the gross values of $\frac{\text{e.t.}}{\text{E.T.}}$.

Effect of Cavitation on Revolutions

In estimating the revolutions where cavitation occurs, the effect of cavitation is exactly the same as that of "thrust deduction." While the power increases, the revolutions increase with it as in "thrust deduction" so that in the equations for apparent slip

$$s=S\frac{K\times \text{i.h.p.}_p\times V^v}{\text{I.H.P.}\times v^v} \quad \text{and} \quad S\frac{K\times V^v}{10^Z\times v^v},$$

the value of i.h.p.$_p$ corrected for cavitation $=M\times$i.h.p.$_p$, and the value Z_1 instead of Z must be used.

On Fig. 10, are shown the values of power and speed plotted on revolutions as abscissas, while on Fig. 9, are shown the values of $\frac{v}{V}$ plotted on $\frac{\text{e.h.p.}}{\text{E.H.P.}}$ as abscissas, while again on Fig. 10, are shown the points where $\frac{\text{e.t.}}{\text{E.T.}}$, equals 1.0, 1.1, 1.15 and 1.225. Attention is called to the erratic character of the *Perry's* curves of power and speed after passing this latter value of $\frac{\text{e.t.}}{\text{E.T.}}$.

The indications from Fig. 9, where $\frac{\text{e.t.}}{\text{E.T.}}=1.15$ coincides with very moderate vibrations, 1.1, to light vibrations, and 1.225 to moderately heavy vibrations were exactly realized on trial.

1. **Effect of Change of Load on Cavitation.** Taking the case of the *Perry* as shown on Fig. 9, it is seen that $\frac{v}{V}$ crosses the line of $\frac{\text{e.t.}}{\text{E.T.}}=1$ at a value of $\frac{v}{V}=.96$ and of $\frac{\text{e.h.p.}}{\text{E.H.P.}}=.96$. Now suppose the load on the vessel be decreased so that for the same effective horse-power the speed be considerably increased. The immediate effect upon the performance is to raise the curve of $\frac{v}{V}$ so that it crosses the line of $\frac{\text{e.t.}}{\text{E.T.}}=1$, at a higher value of $\frac{\text{e.h.p.}}{\text{E.H.P}}$

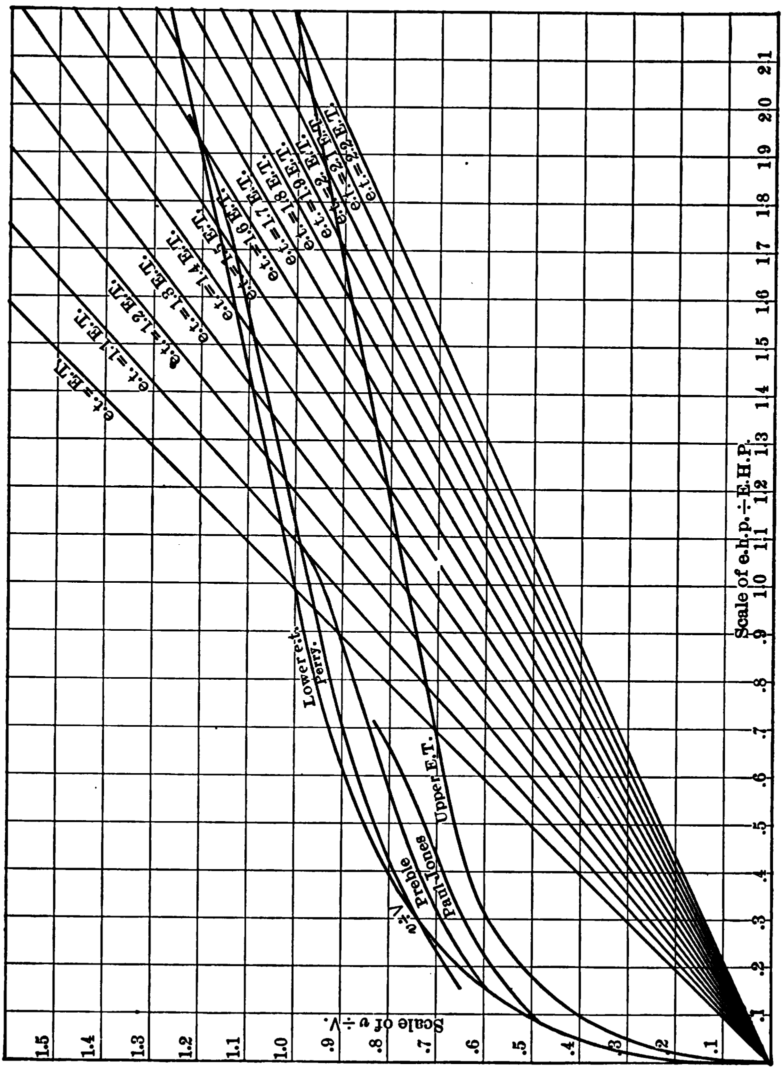


FIG. 9.—Curves of $v \div V$ and e.h.p. ÷ E.H.P. for Destroyers *Paul Jones, Perry* and *Preble.*

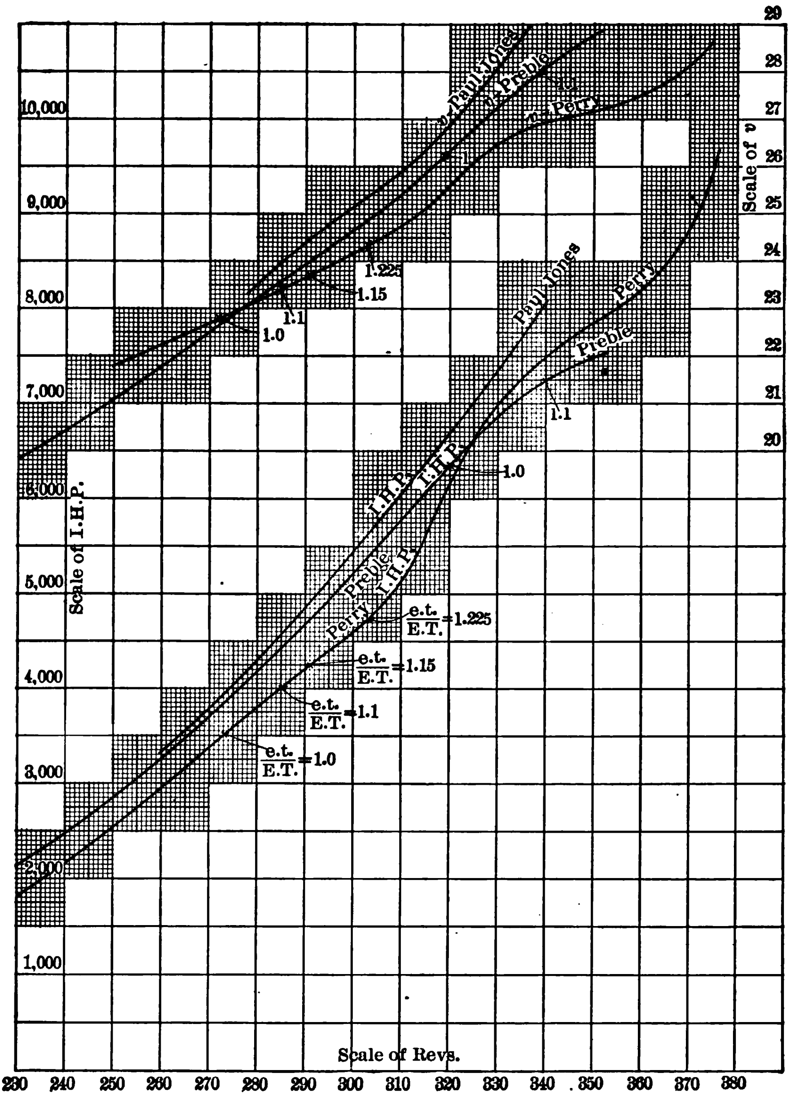


FIG. 10.—Curves of I.H.P.-revolutions and v-revolutions, Destroyers *Paul Jones*, *Perry* and *Preble*.

than before and the entry into the cavitating range is delayed. Should the ship be loaded heavier than at first, the opposite effect occurs, cavitation is produced earlier.

"*To lighten the load on a vessel with a given propeller delays cavitation while to increase the load expedites it.*"

2. **Effect of Change of Projected Area Ratio of the Propeller on Cavitation.** The results accompanying change in projected area ratio are shown very clearly by the performances of the *Paul Jones*, *Perry* and *Preble*.

The *Perry* with a projected area ratio of .275 is on the verge of cavitation with 3500 I.H.P.; with a projected area ratio of .358, the propellers of the *Preble* do not reach the verge until 6300 I.H.P. is being developed. The *Paul Jones*, with a projected area ratio of .43, has not even approached the verge.

"*To increase the projected area ratio of a propeller, pitch and diameter remaining constant, produces a delay in cavitation and in dispersal of the thrust column while to decrease the projected area ratio expedites them.*"

3. **Effect of Change of Pitch, Diameter and Developed Area of the Propeller Remaining Constant, on Production of Cavitation.** To increase the pitch under these conditions reduces the projected area ratio and the effect is similar to that caused by a reduction of projected area only, although the effect is more intensive, as it also lowers the value of $v \div V$ and brings the propeller much closer to cavitation. To lower the pitch produces the opposite effect, therefore, generally speaking,—

"*To increase the pitch of a given propeller tends to expedite cavitation and dispersal of the thrust column while to decrease the pitch tends to delay them.*"

4. **Effect of Reduction of Diameter, the Pitch Remaining Constant, on the Production of Cavitation.** The general effect of such a change is to lower all the Basic conditions of the propeller but the Basic E.H.P. will be lowered more rapidly than the Basic V so that while the factor $\frac{v}{V}$ becomes higher, the factor $\frac{\text{e.h.p.}}{\text{E.H.P.}}$ has increased more rapidly than $\frac{v}{V}$ and the effect, there-

fore, brings the propeller to the verge of cavitation earlier than in its first condition, therefore,—

"*To decrease the diameter of a propeller, the pitch remaining constant, tends to produce earlier cavitation, and dispersal of the thrust column.*"

5. **Effect of "Thrust Deduction" upon the Production of Cavitation.** It has already been pointed out that the value of the power factor Z for any condition where "thrust deduction" does not exist is given by the equation

$$Z_1 = \log \text{I.H.P.} - \log \text{I.H.P.}_p,$$

while should a "thrust deduction" factor K be introduced, the value of Z becomes

$Z_2 = \log\ \text{I.H.P.} - \log\ (K\ \text{I.H.P.}_p) = \log\ \text{I.H.P.} - \log\ \text{I.H.P.}_a$ and the value of Z_2 being less than the value of Z_1, the value of the gross effective horse-power corresponding to Z_2 will be greater than that corresponding to Z_1 and $\left(\frac{\text{e.h.p.}}{\text{E.H.P.}}\right)_2$ will be greater than $\left(\frac{\text{e.h.p.}}{\text{E.H.P.}}\right)_1$, while the values $\frac{v}{V}$ will be the same in both cases. The introduction of K, therefore, results in shifting the curve of $\frac{v}{V}$ horizontally to the right and causing it to intersect the line of $\frac{\text{e.t.}}{\text{E.T.}} = 1$, at a point corresponding to a reduced value of $\frac{v}{V}$ and of $\frac{\text{e.h.p.}}{\text{E.H.P.}}$ below those corresponding to the intersection when no thrust deduction existed, therefore,—

The existence of thrust deduction in addition to increasing the power necessary for propulsion, reduces the speed and net effective thrust at which cavitation will occur.

6. **Effect of "Wake Gain" upon the Production of Cavitation.** The effect of "wake gain" upon speed is the same as that of decrease in resistance. While the effective thrust e.t.÷E.T. is obtained from the model tank curve of speed, e.h.p., the actual speed, due to the wake gain, at which this value of e.t. occurs will be considerably higher than the tank speed, therefore,—

"*Where the hull of a vessel is of such underwater form as to produce a heavy wake, the speed at which cavitation and dispersal of the thrust column occurs will be higher than if no wake existed, on account of the 'wake gain.'*"

7. **Effect of Insufficient Tip Clearance between Propeller and Hull on the Production of Cavitation.** Experience and the analysis of trials of numerous vessels lead to the conclusion that—

"*Where insufficient tip clearance exists between the propeller and the hull, increases in effective horse-power and speed of vessel are accompanied by a gradual increase in the thrust deduction, which latter increase produces earlier cavitation.*"

8. **Effect of Blade Sections on the Production of Cavitation.** Where blade sections are very thick in proportion to their width but their bounding lines are of such form as to give a free flow of water around the section with no tendency to form eddies, "*the abnormal thickness produces an actual pitch considerably greater than the nominal pitch and thus tends to expedite cavitation.*"

Where blade sections are very thick and their bounding lines of such form as to produce eddying of the water at moderate to high revolutions, *the thrust per revolution at the lower revolutions will be increased slightly due to the higher actual pitch produced by the thick section and the power required per revolution will increase in greater proportion than the effective thrust. As the blade speeds increase eddies begin to form and this formation of eddies is accompanied by a still further exaggeration of power and all of the phenomena of cavitation, and this will occur at lower thrusts and speeds than would be the case where the sections were normally fine.*

CHAPTER IX

DESIGN OF THE PROPELLER

Computations for Pitch, Diameter, Projected Area Ratio and Propulsive Efficiency

In computing the principal characteristics of a propeller, these being the pitch, diameter and projected area ratio, the following factors must be considered:

1. The form of the after submerged body of the hull of the vessel to be propelled.

2. The position of the propeller relative to the hull.

3. The effect of the hull lines and position of the propeller in modifying propulsive efficiency.

4. The resistance of the hull to motion through the water at any given speed.

These four points are covered by Sheets 17, 18, and 19 and in some cases by the model tank from which the curves of effective (tow-rope) horse-power are obtained.

In other cases the model tank curves are missing, the tow-rope power is estimated and either this estimate or the estimated I.H.P. or S.H.P. for the speed desired is supplied.

The problems facing the designers of propellers may, therefore, be divided into two classes—

A. Problems of Sufficient Data. In such problems full data of the hull together with the model tank curves of effective horse-powers are provided.

B. Problems of Insufficient Data. In these problems full hull data may be and usually is provided but either an estimate of the effective or of the indicated or shaft horse-power necessary for the desired speed of ships is provided.

These two classes of problems may each be sub-divided into:

C. Problems of Basic conditions (Full Diameter).

D. Problems of reduced load (Reduced Diameter).

In D, the reduction of load may be either positive or negative, that is, the propeller may be designed to deliver less than the Basic condition of E.H.P. or it may be designed to deliver a load greater than the Basic condition of E.H.P., while the designed speed v may be greater or less or equal to the Basic speed V.

PROBLEM A. SUBDIVISION C

Form for Computation

THREE-BLADED PROPELLER

(1) P.A. ÷ D.A. = Different abscissa values taken from Sheet 20.

(2) T.S. = Tip-speeds corresponding to each value of P.A. ÷ D.A. used. Sheet 20.

(3) Slip B.C. = Slip Block Coefficient of vessel. Obtained from Sheet 17.

(4) $1-S$ = 1 − Apparent slip for P.A. ÷ D.A. and Slip B.C. Sheet 20.

(5) I.T.$_D$ = Indicated thrust per square inch of disc area for each value of P.A. ÷ D.A. Sheet 20.

(6) E.H.P. = Effective (tow-rope) horse-power for desired speed. Obtained from model tank curve and includes all appendages.

(7) P.C. = Propulsive coefficient for P.A. ÷ D.A. Sheet 20.

(8) I.H.P. = E.H.P. ÷ P.C. = Indicated horse-power required to deliver E.H.P., without "thrust deduction."

(9) K = Thrust deduction factor for Slip B.C. and for type of vessel and location of propeller. Sheet 19.

(10) K I.H.P. = Total indicated horse-power required.

(11) V = Desired Speed for which E.H.P. is necessary.

(12) $P\times R$ = (V (Knots) × 101.33 or V (Miles) × 88) ÷ ($1-S$) = Pitch × Revs.

(13) D $=\sqrt{(291.8\times \text{I.H.P.})\div(\text{I.T.}_D\times P\times R)}$ = Diameter of Propeller.

(14) P $=(\pi D\times P\times R)\div\text{T.S.}$ = Pitch of the propeller.

(15) R $=\text{T.S.}\div\pi D$ = Revolutions of the propeller.

Should the propeller be a four-bladed one, P.A. ÷ D.A. = $\frac{3}{4}$ the total projected area ratio. The data (2), (4), (5), are taken from Sheet 20 for P.A. ÷ D.A., while the value of P.C. (7) is taken for the full value of the projected area ratio.

The value of D becomes

$$D=\sqrt{(252.41\times \text{I.H.P.})\div(\text{I.T.}_D\times P\times R)}.$$

Should the propeller be a two-bladed one the data (2), (4), (5) are taken for $\frac{3}{2}$ the actual projected area ratio while P.C. is, as before, taken for the actual. The equation for diameter becomes

$$D=\sqrt{(389\times \text{I.H.P.})\div(\text{I.T.}_D\times P\times R)}.$$

In illustrating the above type of problem, the effect of change in speed due to change in resistance and also the effect of an error in the Slip B.C., will be shown.

Problem 25

Statement: Hull Slip B.C. = .5. E.H.P. = 1000. Single screw. The vessel is so loaded at first that a speed of 20 knots an hour requires the above value of E.H.P. Later the vessel is so lightened that a speed of 35 knots can be made with this same E.H.P. Find the diameter, projected area ratio and pitch of the propellers for the two conditions, the desired revolutions being assumed in each case as 600 per minute.

SOLUTION

P.A. ÷ D.A.	.2	.3	.4	.5	.55	.6
T.S.	4200	6650	8580	10550	11830	13550
$1-S$	.884	.88	.869	.849	.832	.807
I.T.$_D$	1.88	3.74	6	8.74	10.3	11.95
E.H.P.	1000	1000	1000	1000	1000	1000
P.C.	.709	.682	.619	.554	.526	.525
I.H.P.	1411	1466	1616	1805	1901	1905
K	1	1	1	1	1	1
V	20	20	20	20	20	20
$P \times R$	2293	2303	2332	2387	2436	2511
D (Feet)	9.772	7.048	5.804	5.025	4.702	4.304
P (Feet)	16.76	7.668	4.956	3.572	3.042	2.506
Revs.	136.8	300.3	470.5	668.3	800.8	1002
V	35	35	35	35	35	35
$P \times R$	4011	4030	4081	4177	4263	4395
D	7.338	5.328	4.388	3.798	3.555	3.253
P	22.17	10.15	6.557	4.725	4.024	3.315
R	181	397.3	622.5	884.1	1059	1326

Plotting these results as shown on Fig. 11, the following propellers are obtained for the two conditions:

V	20	35	E.H.P. constant,
D	5.25	4.42	Increase in Speed
P	3.92	6.85	Decreases Diameter
P.A. ÷ D.A.	.47	.38	Increases Pitch
Blades	3	3	Decreases P.A. ÷ D.A.
I.H.P.$_d$ = I.H.P.$_p$	1743	1600	
P.C.	.5737	.625	Increases Eff. of Prop.
E.H.P.	1000	1000	
R	600	600	

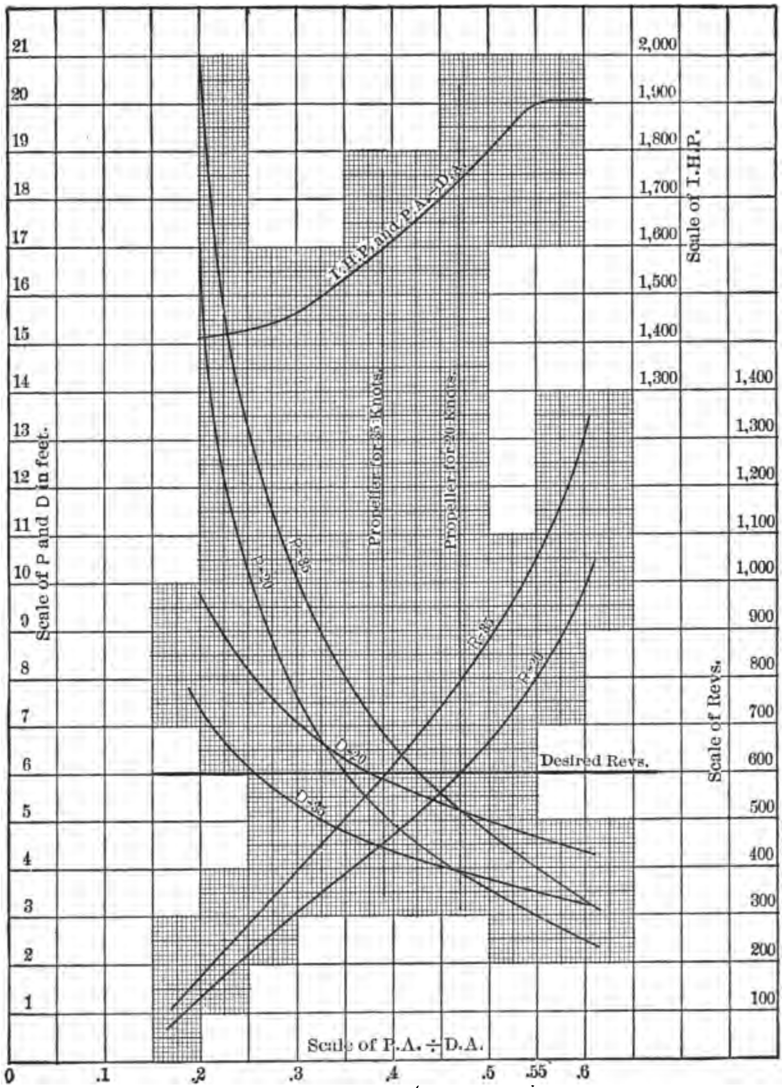


Fig. 11.—Curves of I.H.P., *D*, *P*, and *R*, on P.A.÷D.A. as Abscissas for Full Diameter, Basic Condition Propellers at 20 Knots and at 35 Knots.

Should thrust deduction exist, that is, should K be greater than unity, the actual power required will be

$$K \times \text{I.H.P.}_v = \text{I.H.P.}_d,$$

while the revolutions would be obtained as follows:

$$\text{Apparent Slip} = s = KS,$$

$$\text{Revs.} = R \times \frac{1-S}{1-KS}.$$

These corrections apply both for values of K exceeding unity and below unity, that is for " thrust deduction " and for " wake gain."

Should an error have been made in the estimate of Slip B.C., the following analysis will indicate the effect on the actual performance of the propeller:

Taking the 20-knot condition, but suppose the correct slip B.C. to be .4 instead of .5 as used in the computation.

P.A.÷D.A.	.47	
Blades	3	
D	5.25	
P	3.92	
T.S.	9900	Sheet 20
R	600	
$P \times R$	2352	
Slip B.C.	.4	
$1-S$	.837	Sheet 20
V	19.43	
$v \div V$	1.029	
I.T._D	7.9	Sheet 20
I.H.P	1743	Sheet 20
P.C.	.5737	
E.H.P	1000	
$\log A_V = \log (V^y)$	3.85	Sheet 21
$\log A_v = \log (v^y)$	3.89	Sheet 21
$s = S \dfrac{\text{I.H.P.}V^y}{\text{I.H.P.}v^y}$	.1487	
$R_d = R\left(\dfrac{v}{V} \times \dfrac{1-S}{1-s}\right)$	607.1	

Such an error produces no change in the power required for the speed unless the value of $v \div V$ for the load factor becomes lower than that corresponding to $E.T.$ for this same load factor, on Sheet 22, or the change be such as to produce a change in the value of K, but increases the revolutions above those calculated. Should the slip B.C. be higher than that used, the actual revolutions will be lower than the estimated.

Problems A. Subdivision D

Such problems are those which are encountered when the suitable propellers for vessels of low to moderate speeds, revolutions and power are being sought. With such conditions the Basic conditions of design are far in excess of the actual conditions, and the actual data of desired performance must be so handled as to bring it up to the Basic conditions before the work of calculation can be undertaken.

Method of Design

Assumption of Diameter, Load and Speed Factors to find Projected Area Ratio, Pitch, Revolutions, Power on and Effective Power delivered by the Propeller.

By inspection of Sheet 22, it willl be seen that there are shown two limiting curves of effective thrust and several curves of thrust marked for various types of vessels. The ordinates of these curves are values of $v \div V$ while the abscissas are values of e.h.p. $\div$ E.H.P.

What occurs above the upper limiting curve is not known, but between the limits the efficiency of any propeller for any particular value of e.h.p. $\div$ E.H.P. remains practically constant while below the lower limiting curve the efficiency falls very rapidly as the value of $v \div V$ decreases.

The intermediate curve is derived from the performances of some very successful vessels and is given as a guide to locate the

desired propeller for any given type of vessel. For instance, heavy and full-bodied merchant ships should be located between the upper curve and the second one from it in order to hold sufficient range to take care of deep-load and adverse weather conditions. Very fine vessels such as destroyers and speed boats when designed for high power and speed fall in this same range. Hydroplanes may, and usually do, plot far above the upper curve. Vessels of nearly constant condition of loading and of comparatively low revolutions for the power, should plot on or near the second curve from the upper limit one, that is vessels such as the U.S.S. *Texas*, *Delaware*, *Pennsylvania*, with revolutions from 125 to 220 for powers ranging from 25,000 on two shafts to 30,000 on four shafts, all plot in this range, while the *Arkansas*, with 330 revolutions for 28,000 S.H.P. on four shafts, plots below the intermediate curve, and the *Utah* and *Florida* for the same revolutions and power plot almost exactly on the lower curve or curve of critical thrusts.

It might be inferred from Sheet 22, that any load factor can be used in the design of the propeller without regard to either the slip-block coefficient of the vessel or to the speed of ship, but such is not the case. The three factors tie together and for a vessel having a given slip block coefficient and designed for a certain given speed there exists a load factor for design which must not be exceeded if estimated propulsive efficiencies are to be realized. The curves of approximate maximum and minimum values of e.h.p. ÷ E.H.P. for different slip block coefficients from .2 to 1.0, varying by .1, are given on Sheet 22B, of which the abscissas are speeds, v, and ordinates, e.h.p. ÷ E.H.P.

In selecting values of e.h.p. ÷ E.H.P. to use in the calculations, they should usually be taken between these maximum and minimum limits for the designed speed and slip block coefficient, but the maximum values may be exceeded by fully 25 per cent with safety.

The equations for finding the diameter of the propeller have already been given, but they will be given again and also an additional one in terms of effective horse-power and effective thrust per square inch of projected area ratio.

These equations are

For 2 Blades:

$$D=\sqrt{\frac{389\times \text{I.H.P.}}{\text{I.T.}_D\times P\times R}}=\sqrt{\frac{3.84\times \text{I.H.P.}\times(1-S)}{\text{I.T.}_D\times V}}$$

$$=\sqrt{\frac{3.84\times \text{E.H.P.}}{\frac{\text{P.A.}}{\text{D.A.}}\times \text{E.T.}_p\times V}};$$

For 3 Blades:

$$D=\sqrt{\frac{298.1\times \text{I.H.P.}}{\text{I.T.}_D\times P\times R}}=\sqrt{\frac{2.88\times \text{I.H.P.}\times(1-S)}{\text{I.T.}_D\times V}}$$

$$=\sqrt{\frac{2.88\times \text{E.H.P.}}{\frac{\text{P.A.}}{\text{D.A.}}\times \text{E.T.}_p\times V}};$$

For 4 Blades:

$$D=\sqrt{\frac{252.41\times \text{I.H.P.}}{\text{I.T.}_D\times P\times R}}=\sqrt{\frac{2.491\times \text{I.H.P.}\times(1-S)}{\text{I.T.}_D\times V}}$$

$$=\sqrt{\frac{2.491\times \text{E.H.P.}}{\frac{\text{P.A.}}{\text{D.A.}}\times \text{E.T.}_p\times V}};$$

in which $\frac{\text{P.A.}}{\text{D.A.}}$ equals $\frac{3}{2}$ times for the two-bladed, equals for the three-bladed and equals $\frac{3}{4}$ times for the four-bladed, the projected area ratio of the propeller, E.T._p, I.T._D, and $(1-S)$ being those corresponding to P.A.$\div$D.A. of the equivalent three-bladed propeller.

From the above equations, D, V and I.H.P. or D, V and E.H.P. being known, the value of $\text{I.T.}_D\div(1-S)$ in the first case and of $(\text{P.A.}\div\text{D.A.})\times\text{E.T.}_p$, in the second case, can be obtained:

$$\text{I.T.}_D\div(1-S)=\frac{3.84\ \text{I.H.P.}}{D^2\times V},\ 2\text{ blades};$$

$$=\frac{2.88\ \text{I.H.P.}}{D^2\times V},\ 3\text{ blades};\quad =\frac{2.491\ \text{I.H.P.}}{D^2\times V},\ 4\text{ blades};$$

and

$$(\text{P.A.} \div \text{D.A.}) \times \text{E.T.}_{p} = \frac{3.84 \text{ E.H.P.}}{\text{D}^2 \times V}, \text{ 2 blades;}$$

$$= \frac{2.88 \text{ E.H.P.}}{D^2 \times V}, \text{ 3 blades;}$$

$$= \frac{2.491 \text{ E.H.P.}}{D^2 \times V}. \text{ 4 blades.}$$

Values of $\text{I.T.}_D \div (1-S)$ and of $(\text{P.A.} \div \text{D.A}) \times \text{E.T.}_p$ are given in the accompanying tables for different values of slip-block coefficient and of projected area ratios and are plotted as curves on sheets 23 and 24.

Having obtained the values of P.A.÷D.A. from the values of $\text{I.T.}_D \div (1-S)$ or of $(\text{P.A.} \div \text{D.A.}) \times \text{E.T.}_p$, reducing them to $\frac{4}{3}$ P.A.÷D.A. for four-bladed and to $\frac{2}{3}$ P.A.÷D.A. for two-bladed propellers, the propulsive coefficient corresponding to these total projected area ratios, the tip-speeds and $(1-S)$ values corresponding to the basic P.A.÷D.A. can all be obtained from Design Sheet 20, and the problem solved, following the form given on page 156:

TABLE OF $\frac{\text{P.A.}}{\text{D.A.}} \times \text{E.T.}_p$

Slip B.C.	P.A. ÷ D.A.									
	.2	.25	.3	.35	.4	.45	.5	.55	.6	.65
.9	1.367	2.014	2.651	3.287	3.905	4.568	5.19	5.891	7.049	8.172
.85	1.383	2.037	2.682	3.322	3.951	4.621	5.263	5.969	7.146	8.249
.8	1.399	2.061	2.711	3.361	3.989	4.667	5.309	6.029	7.211	8.368
.75	1.417	2.087	2.74	3.405	4.037	4.733	5.38	6.125	7.312	8.471
.7	1.433	2.112	2.773	3.446	4.090	4.791	5.459	6.203	7.407	8.555
.65	1.452	2.138	2.806	3.488	4.141	4.845	5.527	6.282	7.496	8.682
.6	1.47	2.164	2.84	3.523	4.187	4.906	5.579	6.356	7.595	8.792
.55	1.485	2.188	2.872	3.571	4.240	4.951	5.657	6.431	7.689	8.905
.5	1.498	2.208	2.902	3.607	4.273	5.008	5.703	6.501	7.774	9.021
.45	1.514	2.231	2.932	3.645	4.319	5.067	5.764	6.58	7.846	9.127
.4	1.526	2.251	2.966	3.683	4.369	5.128	5.834	6.653	7.952	9.236
.35	1.549	2.28	3.001	3.723	4.422	5.189	5.905	6.728	8.044	9.348
.3	1.565	2.31	3.037	3.772	4.475	5.253	5.978	6.813	8.148	9.476

TABLE OF I.T.$_D$÷(1−S)

Slip B.C.	P.A. ÷ D.A.									
	.2	.25	.3	.35	4.	.45	.5	.55	.6	.65
.9	1.928	2.865	3.888	5.042	6.309	7.798	9.368	11.2	13.43	15.57
.85	1.95	2.898	3.933	5.095	6.383	7.886	9.50	11.35	13.61	15.71
.8	1.973	2.931	3.974	5.155	6.445	7.963	9.584	11.46	13.74	15.94
.75	1.998	2.969	4.017	5.222	6.522	8.077	9.711	11.65	13.93	16.14
.7	2.022	3.004	4.065	5.285	6.608	8.176	9.854	11.79	14.11	16.30
.65	2.048	3.041	4.115	5.349	6.689	8.268	9.977	11.94	14.28	16.54
.6	2.073	3.078	4.165	5.415	6.765	8.372	10.07	12.08	14.47	16.75
.55	2.094	3.112	4.212	5.476	6.849	8.448	10.21	12.23	14.65	16.96
.5	2.113	3.141	4.255	5.533	6.903	8.547	10.3	12.36	14.81	17.18
.45	2.135	3.173	4.299	5.59	6.977	8.647	10.41	12.51	14.95	17.39
.4	2.152	3.202	4.349	5.649	7.059	8.75	10.53	12.65	15.15	17.59
.35	2.185	3.244	4.40	5.709	7.143	8.856	10.66	12.79	15.32	17.81
.3	2.208	3.286	4.452	5.785	7.229	8.964	10.79	12.95	15.52	18.05

Suppose all revolutions obtained are higher than those desired, while the projected area ratio has decreased to and below .25 for the three-blade basic propeller. Investigation of Sheet 22 reveals that so long as the ratio of e.t. to E.T. remains constant the projected area ratio will remain constant but that as we pass down this line of constant ratio of e.t. to E.T., with constant diameter of propeller, the pitch of the propeller will increase and the revolutions decrease. Therefore, taking P.A.÷D.A. constant as derived from the first step, and either equal to .25 or to that value of P.A.÷D.A. which by inspection will result in a good ratio of $P \div D$ without bringing the propeller to plot on Sheet 22, too close to the curve of critical thrusts as to so plot, in the cases of vessels subjected to great variation of load conditions, might bring the thrusts greater than the critical thrusts under heavy load conditions and an excessive falling off in propulsive efficiency would result.

The form for the computation follows on page 157.

SCREW PROPELLERS

FORM FOR COMPUTATION. e.h.p., SPEED AND REVOLUTIONS FIXED. $I.H.P._d$ OR $S.H.P._d$ UNKNOWN.—FIRST STEP

D											
e.h.p. ÷ E.H.P. (assumed)											
e.h.p.											
$\text{E.H.P.} = \text{e.h.p.} \div \dfrac{\text{e.h.p.}}{\text{E.H.P.}}$											
v											
$v \div V$ for $\dfrac{\text{e.h.p.}}{\text{E.H.P.}} \left(\begin{matrix}\text{above}\\ \text{Crit. Thr.}\end{matrix}\right)$ (Sheet 22)											
V											
$(\text{P.A.} \div \text{D.A.}) \times \text{E.T.}_p = \dfrac{C \times \text{E.H.P.}}{D^2 \times V}$											

NOTE: Values of $C = 3.84$, 2.88, 2.491, for 2, 3, and 4 blades.

P.A. ÷ D.A. for $\dfrac{\text{P.A.}}{\text{D.A.}} \times \text{E.T.}_p$ (from Sheet 24)											
$\frac{2}{3}$ P.A. ÷ D.A. for 2 blades											
$\frac{4}{3}$ P.A. ÷ D.A. for 4 blades											
P.C. for total $\dfrac{\text{P.A.}}{\text{D.A.}}$											
I.H.P. = E.H.P. ÷ P.C.											
Z for e.h.p. ÷ E.H.P. (Sheet 21)											
I.H.P._p											
Slip B.C.											
K for Slip B.C.											
I.H.P._d											
$\text{S.H.P.}_d = \text{I.H.P.}_d \times .92$											
$1 - S$ for P.A. ÷ D.A.											
T.S. for P.A. ÷ D.A.											
$P = \dfrac{\pi \times D \times V \times 101.33}{\text{T.S.} \times (1 - S)}$											

Now suppose that the value of $v \div V$ were such as to plot on Sheet 22 for the assumed values of e.h.p ÷ E.H.P., below the curve of critical thrusts, then for each of the assumed values of e.h.p. ÷ E.H.P. we have the following values of

$(E.T. \div \text{e.t.})^x$											
$K^1 = K \div \left(\dfrac{E.T.}{\text{e.t.}}\right)^x$											
$\text{I.H.P.}_d = \text{I.H.P.}_p \times K^1$											

TO FIND REVOLUTIONS

$\log A_V$											
$\log A_v$											
$s = S \dfrac{\text{I.H.P.}_d \times A_V}{\text{I.H.P.} \times A_v}$											
$R_d = \dfrac{v \times 101.33}{P \times (1 - s)}$											

FORM FOR SECOND STEP

Total Proj. Area Ratio (Constant)	....	Constant		
P.A.÷D.A.= $\left\{\begin{array}{l}\frac{3}{2}\text{ Total for 2 blades}\\ \text{Total for 3 blades}\\ \frac{3}{4}\text{ Total for 4 blades}\end{array}\right\}$	....	Constant		
P.C. for Total Proj. Area Ratio	....	Constant		
e.t.÷E.T. (constant)				
e.h.p.÷E.H.P. (variable)				
$v \div V$ for $\frac{\text{e.t.}}{\text{E.T.}}$ and $\frac{\text{e.h.p.}}{\text{E.H.P.}}$				
v (designed speed, knots)				
e.h.p. (designed eff. horse-power)				
$V = v \div (v \div V)$				
E.H.P. = e.h.p. ÷ (e.h.p. ÷ E.H.P.)				
I.H.P. = E.H.P. ÷ P.C				
S.H.P. = I.H.P. × .92				
T.S. = Tip-speed for P.A. ÷ D.A				
Slip B.C. (as in First Step)				
$1-S$ for P.A.÷D.A. and Slip B.C.				
D = Diameter (Fixed by First Step)				
$P = \frac{101.33 \times \pi \times V \times D}{\text{T.S.} \times (1-S)} = \frac{318.3 \times V \times D}{\text{T.S.} \times (1-S)}$				
K as in First Step				
Z for e.h.p.÷E.H.P.				
$\text{I.H.P.}_p, \text{S.H.P.}_p = (\text{I.H.P., S.H.P.}) \div 10^Z$				
$\text{I.H.P.}_d, \text{S.H.P.}_d = K \times (\text{I.H.P.}_p, \text{S.H.P.}_p)$				
Log A_V (for V Sheet 21)				
Log A_v (for v, Sheet 21)				
$s = S\,\frac{\text{I.H.P.}_d \text{ or S.H.P.}_d \times A_V}{\text{I.H.P. or S.H.P.} \times A_v}$				
$R_d = \frac{v \times 101.33}{P \times (1-s)}$				

SCREW PROPELLERS

Insufficient Data

FORM FOR COMPUTATION—I.H.P.$_d$ OR S.H.P.$_d$ AND REVOLUTIONS AND SPEED FIXED—e.h.p. FOR v UNKNOWN

D	Max. Carried.	2d—Less than Max.	3d—Less than 2d.
e.h.p. ÷ E.H.P. (assumed)			
Z for $\frac{\text{e.h.p.}}{\text{E.H.P.}}$			
I.H.P.$_d$ or S.H.P.$_d$	Constant	Constant	Constant
Slip B.C.	Constant	Constant	Constant
K for S.B.C	Constant	Constant	Constant
I.H.P.$_p$ or S.H.P.$_p$	Constant	Constant	Constant
S.H.P.			
I.H.P.			
v (Des. Speed)	Constant	Constant	Constant
$v \div V$ for $\frac{\text{e.h.p.}}{\text{E.H.P.}} \left(\begin{matrix}\text{above}\\ \text{Crit. Thr.}\end{matrix}\right)$			
$V = v \div \left(\frac{v}{V}\right)$			
$\text{I.T.}_D \div (1-S) = \frac{C \times \text{I.H.P.}}{D^2 \times V}$			
($C=3.84$ for 2 blade,	2.88 for 3 blade	and 2.491 for 4	blade).
P.A. ÷ D.A. for I.T.$_D \div (1-S)$ $\frac{2}{3}$ P.A. ÷ D.A. (for 2-blade) $\frac{4}{3}$ P.A. ÷ D.A. (for 4-blad)	Total Projected	Area Ratios.	
P.C. for Total $\frac{\text{P.A.}}{\text{D.A.}}$			
E.H.P. = I.H.P. × P.C.			
e.h.p. = E.H.P. × $\frac{\text{e.h.p.}}{\text{E.H.P.}}$			
$1-S$ for (P.A. ÷ D.A.)			
T.S. for (P.A. ÷ D.A.)			
$P = \frac{\pi D \times V \times 101.33}{\text{T.S.} \times (1-S)}$			

TO FIND REVOLUTIONS

$\log A_V$			
$\log A_v$	Constant	Constant	Constant
$s = S \frac{\text{I.H.P.}_d \times A_V}{\text{I.H.P.} \times A_v}$			
$R_d = \frac{v \times 101.33}{P \times (1-s)}$			

Should there be a possibility of the speed being reduced to v_1 while I.H.P.$_d$ or S.H.P.$_d$ remained constant, the value of v_1 being

such that there is danger that the values e.h.p.$_1$ ÷ E.H.P. and $v_1 \div V$ will plot below the curve of critical thrusts, it is desired to find the value of e.h.p.$_1$ ÷ E.H.P. and of e.h.p.$_1$ which will be delivered under the new speed condition.

It is necessary to bear in mind that the fundamental equation for apparent slip is

$$s = S \times \frac{\text{I.H.P.}_d \times V^y}{\text{I.H.P.} \times v^y} = S \times \frac{\text{I.H.P.}_d \times A_V}{\text{I.H.P.} \times A_v},$$

while where $v_1 \div V$ falls below the critical thrust curve of Sheet 22,

$$s = S \times \frac{K \times V^y}{10^{Z_1} \times v^y} \times \left(\frac{\text{e.t.}}{E.T.}\right)^x = S \times \frac{K \times A_V}{10^{Z_1} \times A_v} \times \left(\frac{\text{e.t.}}{E.T.}\right)^x.$$

Now these two values of s must be the same, therefore:

$$S \times \frac{\text{I.H.P.}_d \times A_V}{\text{I.H.P.} \times A_v} = S \times \frac{K \times A_V}{10^{Z_1} \times A_v} \times \left(\frac{\text{e.t.}}{E.T.}\right)^x$$

$$\frac{\text{I.H.P.}_d}{\text{I.H.P.}} = \frac{K}{10^{Z_1}} \times \left(\frac{\text{e.t.}}{E.T.}\right)^x,$$

$$\log \text{I.H.P.}_d - \log \text{I.H.P.} = \log K + x \log \left(\frac{\text{e.t.}}{E.T.}\right) - Z_1,$$

$$Z_1 = \log \text{I.H.P.} - \log \text{I.H.P.}_d + \log K + x \log \left(\frac{\text{e.t.}}{E.T.}\right).$$

The value of e.h.p.$_1$ ÷ E.H.P. for this value of Z_1 will be that delivered and

$$\text{e.h.p.}_1 = \text{E.H.P.} \times (\text{e.h.p.}_1 \div \text{E.H.P.}).$$

Should the second step be necessary to obtain the desired revolutions, proceed as in the preceding case.

In choosing the values of e.h.p. ÷ E.H.P. to use in the computations, in no case should the values fall over 25 per cent outside the maximum and minimum limits as given by Sheet 22B for the slip B.C. and the designed speed.

Problem 26.—Full Data. Effective Horse-power Used

Vessel of "Tanker" type. Slip B.C. = .80.

Speed loaded 11 knots. Revs. 90; e.h.p. for designed speed = 1500. Single screw. Draft of vessel in excess of 20 ft. Maximum diameter of propeller that can be carried = 18 ft. Determine characteristics of four-

bladed propeller and shaft horse-power necessary, the propelling engines being of the geared turbine type.

SOLUTION

D	18′	18′	18′	17′	17′	17′	16′	16′	16′
$\frac{\text{e.h.p.}}{\text{E.H.P.}}$	.2	.3	.4	.2	.3	.4	.2	.3	.4
e.h.p.	1500	1500	1500	1500	1500	1500	1500	1500	1500
E.H.P.	7500	5000	3750	7500	5000	3750	7500	5000	3750
v	11	11	11	11	11	11	11	11	11

As the conditions given are for the vessel at full load, it is only necessary to provide for sufficient leeway above the curve of critical thrusts, Sheet 22, to take care of average rough bottom and bad weather, therefore, take the values of $v \div V$ from the curve on Sheet 22, marked "Curve of Maximum Efficiency."

$v \div V$	.575	.662	.73	.575	.662	.73	.575	.662	.73
V	19.13	16.62	15.07	19.13	16.62	15.07	19.13	16.62	15.07

Since the propeller is four-bladed and the e.h.p. is being used, the value

$$(\text{P.A.} \div \text{D.A.}) \times \text{E.T.}_p = \frac{2.491 \times \text{E.H.P.}}{D^2 \times V}. \quad \therefore$$

$\frac{\text{P.A.}}{\text{D.A.}} \times \text{E.T.}_p$	2.813	2.313	1.913	3.154	2.593	2.145	3.552	2.927	2.422
$\frac{\text{P.A.}}{\text{D.A.}}$ (Sheet 24)	.308	.27	.240	.334	.291	.256	.364	.316	.278

The total projected area ratio of the four-bladed screw being $\frac{4}{3} \times$ that of the basic three-bladed one:

$\frac{4}{3}$ P.A. $\div$ D.A.	.412	.360	.320	.444	.388	.340	.484	.420	.372

The basic value of the propulsive coefficient being dependent upon total projected area ratio, we have P.C. for $\frac{4}{3}$ P.A. $\div$ D.A., Sheet 20.

P.C.	.611	.646	.67	.59	.6275	.659	.564	.606	.638
I.H.P. = E.H.P. $\div$ P.C.	11456	7740	5597	11864	7968	5690	12408	8251	5878
S.H.P. = I.H.P. $\times$ .92	10540	7121	5149	10915	7331	5235	11416	7591	5408
Z for $\frac{\text{e.h.p.}}{\text{E.H.P.}}$ (Sheet 21)	.7279	.5445	.4144	.7279	.5445	.4144	.7279	.5445	.4144
$\text{S.H.P.}_p = \text{S.H.P.} \div 10^Z$	1972	2033	1983	2042	2093	2012	2136	2167	2087

As the vessel is single screw of .80 slip block coefficient and is over 20 ft. in draft, the value of the thrust deduction factor K is obtained from the curve $C_3 - C_2$, Sheet 19, and is equal to 1.27. ∴

K	1.27	1.27	1.27	1.27	1.27	1.27	1.27	1.27	1.27
S.H.P.$_d$ = $K\times$S.H.P.$_p$	2505	2581	2519	2594	2657	2561	2713	2752	2651

These values S.H.P.$_d$ are those of the necessary designed powers for the series of propellers obtained.

The basic values of tip-speeds and of $1-$basic apparent slip are obtained from Sheet 20, using the basic P.A. ÷ D.A. values of the basic three-bladed propeller, the values of $1-S$ being taken from the curve of $1-S$ for slip B.C. = .80. ∴

T.S. for $\frac{\text{P.A.}}{\text{D.A.}}$	6800	5980	5250	7330	6450	5650	7820	6990	6190
$1-S$ for $\frac{\text{P.A.}}{\text{D.A.}}$	.941	.944	.947	.939	.942	.945	.935	.940	.943

Now the pitch of the propeller equals $\frac{\pi\times D\times V\times 101.33}{\text{T.S.}\times(1-S)}$. ∴

P	17′.13	16′.87	17′.37	15′.04	14′.8	15′.27	13′.33	12′.88	13′.15

To obtain the revolutions which may be expected from this series of propellers when operating under the designed conditions of speed and effective horse-power, the data is obtained from Sheet 21, where will be found a curve of values of log A_V and v, V being the basic speed as found in the foregoing calculations, and v being the designed speed of 11 knots.

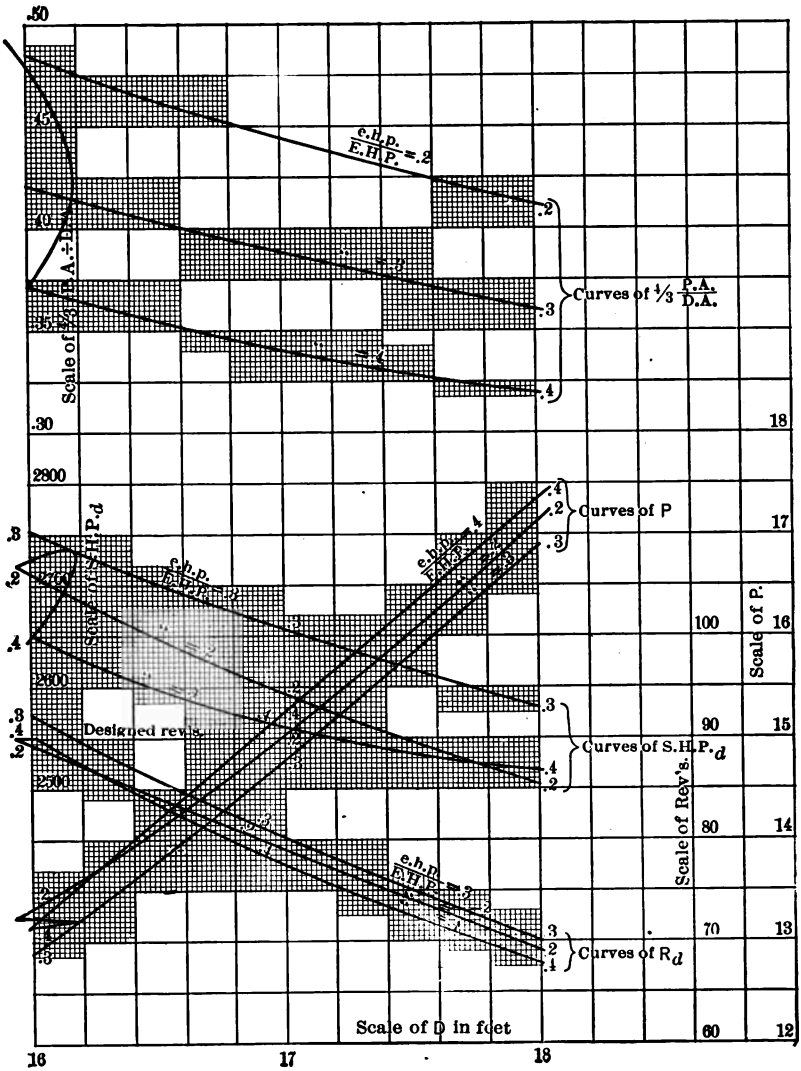
Log A_V	3.83	3.655	3.53	3.83	3.655	3.53	3.83	3.655	3.53
Log A_v	3.21	3.21	3.21	3.21	3.21	3.21	3.21	3.21	3.21

The equation for the apparent slip at the designed speed is

$$s=S\frac{\text{S.H.P.}_d\times A_V}{\text{S.H.P.}\times A_v}. \therefore$$

s	.05845	.05656	.05416	.06043	.05858	.0562	.06439	.06060	.05838
$R_d=\frac{v\times 101.33}{P\times(1-s)}$	69.1	70.05	67.86	78.87	80	77.33	89.4	92.12	90.02

These values of R_d are the revolutions for the series of resultant propellers at a speed of vessel of 11 knots, delivering 1500 e.h.p. with S.H.P.$_d$ shaft horse-power. The derived values of S.H.P.$_d$, P, $\frac{4}{3}\,\frac{\text{P.A.}}{\text{D.A.}}$ and R_d can now be plotted on cross section paper, using values of D as abscissas (Fig. 12) and that propeller giving the desired revolutions, with its diameter, pitch projected area ratio and necessary shaft horse-power can be taken off the curves.



Curves of 1/3 P.A./D.A.
Curves of P
Curves of S.H.P.d
Curves of Rd
Scale of 1/3 P.A. ÷ D.A.
Scale of S.H.P.d
Scale of Rev's.
Scale of P.
Scale of D in feet
Designed rev's.
e.h.p./E.H.P. = .2
e.h.p./E.H.P. = .3
e.h.p./E.H.P. = .4

FIG. 12.

It is seen at once that the 16 ft. diameter propeller, having a projected area ratio of .372 and a pitch of 13.15 ft. will answer the conditions.

However, a more efficient propeller can be obtained by increasing the diameter and the methods for doing this are illustrated in the following calculations, taking the minimum projected area ratio propellers of 17 and 18 ft. diameters and solving, first, with constant $\frac{\text{e.t.}}{\text{E.T.}}$, and second, with $\frac{\text{e.h.p.}}{\text{E.H.P.}}$ constant,

CONSTANT EFFECTIVE THRUST

D	18	18	18	17	17	17
v	11	11	11	11	11	11
e.h.p.	1500	1500	1500	1500	1500	1500
e.t. ÷ E.T.	.55	.55	.55	.55	.55	.55

This value of the ratio between e.t. and E.T. is that existing at the point where e.h.p. ÷ E.H.P. = .4, used in the previous calculations, was taken. Holding this value of e.t. ÷ E.T. constant and increasing $\frac{\text{e.h.p.}}{\text{E.H.P.}}$, with the corresponding increased values of $v \div V$, we have:

e.h.p. ÷ E.H.P.	.45	.5	.55	.45	.5	.55
$v \div V$	.82	.91	1.0	.82	.91	1.0
V	13.42	12.09	11	13.42	12.09	11
E.H.P.	3333	3000	2727	3333	3000	2727
$\frac{\text{P.A.}}{\text{D.A.}} \times \text{E.T.}_p = \frac{2.491 \text{ E.H.P.}}{D^2 \times V}$	1.911	1.908	1.906	2.142	2.139	2.137
P.A. ÷ D.A.	.24	.24	.24	.256	.256	.256
$\frac{4}{3}$ P.A. ÷ D.A.	.32	.32	.32	.34	.34	.34
P.C. for $\frac{4}{3}$ (P.A. ÷ D.A.)	.67	.67	.67	.659	.659	.659
I.H.P.	4975	4478	4071	5058	4552	4139
S.H.P.	4577	4120	3745	4654	4188	3808
Z for $\frac{\text{e.h.p.}}{\text{E.H.P.}}$	.36	.3135	.27	.36	.3153	.27
S.H.P.$_p$	1998	2002	2011	2031	2035	2045
K	1.27	1.27	1.27	1.27	1.27	1.27
S.H.P.$_d$	2538	2542	2554	2580	2584	2597
T.S. for P.A. ÷ D.A.	5250	5250	5250	5650	5650	5650
$1-S$ for P.A. ÷ D.A. and slip B.C. = .8	.947	.947	.947	.945	.945	.945
P	15.46	13.93	12.68	13.6	12.25	11.15
Log A_V	3.38	3.25	3.21	3.38	3.25	3.21
Log A_v	3.21	3.21	3.21	3.21	3.21	3.21
s	.04346	.03586	.03616	.04510	.03721	.03751
R_d	75.37	82.98	91.22	85.85	94.49	103.9

CONSTANT $\frac{\text{e.h.p.}}{\text{E.H.P.}}$

D	18	18	18	17	17	17
v	11	11	11	11	11	11
e.h.p.	1500	1500	1500	1500	1500	1500
e.h.p. ÷ E.H.P.	.4	.4	.4	.4	.4	.4
E.H.P.	3750	3750	3750	3750	3750	3750
$v \div V$	.82	.91	1.0	.82	.91	1.0
V	13.42	12.09	11	13.42	12.09	11
$\frac{\text{P.A.}}{\text{D.A.}} \times \text{E.T.}_p$	2.148	2.385	2.621	2.409	2.674	2.938
P.A. ÷ D.A.	.257	.275	.293	.277	.296	.317
$\frac{4}{3}$ P.A. ÷ D.A.	.344	.368	.392	.372	.396	.424
P.C.	.656	.641	.625	.638	.622	.604
I.H.P.	5717	5850	6000	5878	6029	6209
S.H.P.	5259	5382	5520	5408	5547	5712
Z	.4144	.4144	.4144	.4144	.4144	.4144
S.H.P._p	2026	2073	2126	2083	2136	2200
K	1.27	1.27	1.27	1.27	1.27	1.27
S.H.P._d	2572	2633	2700	2645	2713	2794
T.S.	5550	6100	6500	6150	6570	7000
$1-S$	.946	.944	.942	.943	.942	.940
P	14.65	12.03	10.3	12.52	10.57	9.047
Log A_V	3.38	3.25	3.21	3.38	3.25	3.21
Log A_v	3.21	3.21	3.21	3.21	3.21	3.21
s	.03907	.03003	.02837	.04124	.03111	.02935
R_d	79.2	95.52	111.4	92.41	108.8	126.9

Plotting the results obtained by these last two calculations together with the values obtained for the $\frac{\text{e.h.p.}}{\text{E.H.P.}} = .4$, points of the first calculations, the following propellers are obtained, all for 90 revolutions, delivering 1500 e.h.p. at 11 knots speed of ship.

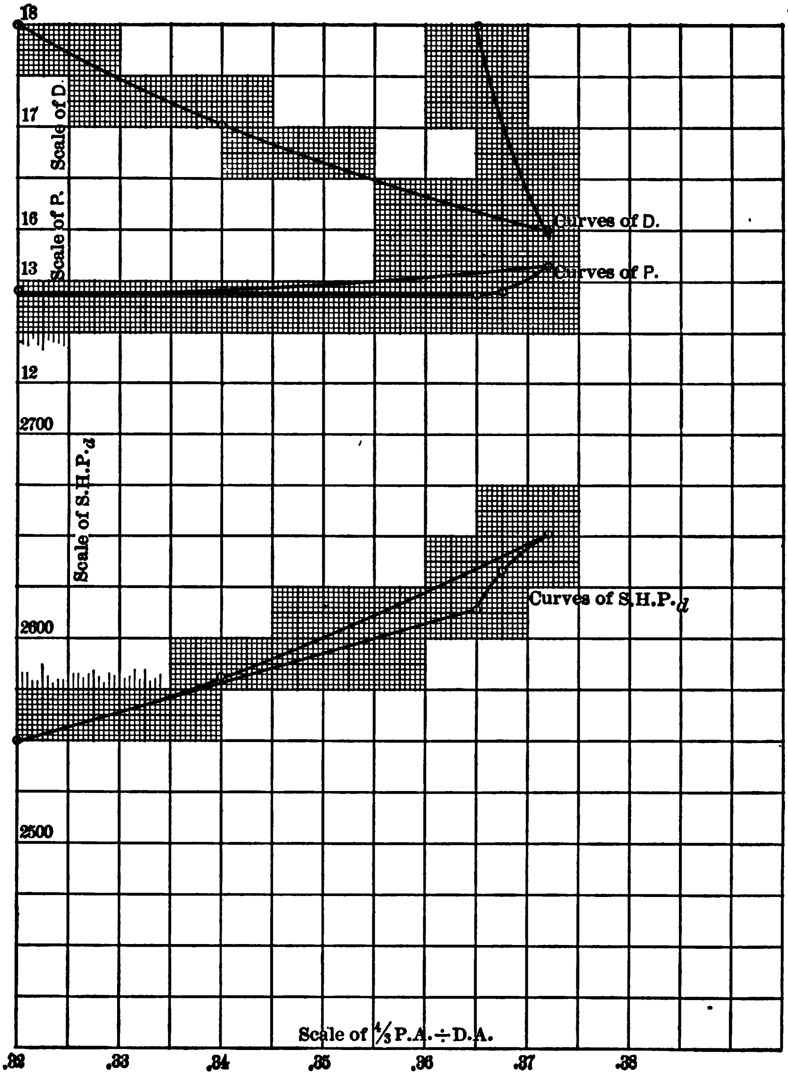

18
17
16
13
12
Scale of D.
Scale of P.
Curves of D.
Curves of P.
2700
2600
2500
Scale of S.H.P.$_d$
Curves of S.H.P.$_d$
Scale of 4/3 P.A. ÷ D.A.
.32
.33
.34
.35
.36
.37
.38

FIG. 13.

CONSTANT $\frac{e.t.}{E.T.}$ CONSTANT $\frac{e.h.p.}{E.H.P.}$

		Constant $\frac{e.t.}{E.T.}$		Constant $\frac{e.h.p.}{E.H.P.}$	
Diam	16′	17′	18′	17′	18′
Pitch	13′.15	12′.9	12′.87	12′.9	12′.87
$\frac{P.A.}{D.A.}$	.372	.34	.32	.3675	.36
S.H.P.	2651	2582	2550	2633	2614
e.h.p.	1500	1500	1500	1500	1500
p.c.	.566	.582	.627	.57	.574
R_d	90	90	90	90	90
v	11	11	11	11	11

By plotting the above characteristics and running cross curves (Fig. 13), an innumerable number of propellers can be obtained with diameters varying from 16 ft. to 18 ft., pitches from 13.15 to 12.87, and total projected area ratios from .32 to .372. The shaft horse-powers required for all of these propellers varies from 2550 to 2651, a difference between the best and the poorest of only about 4 per cent or 100 shaft horse-power.

It is such peculiarities in propeller performances that create so many different opinions as to what is the proper propeller to use for any particular problem, the experiences of the various designers have placed no two in exactly the same position of the zone of design.

However, as the 17-ft. propellers above fall approximately on the upper limit of well-known propeller design territory, it would be well to confine ourselves to this diameter and then the choice narrows to that of the projected area ratio to use.

By inspection of the above table of propellers it will be seen that both propellers given with 17-ft. diameter vary from a projected area ratio of .34 to one of .3675, while the pitch remains constant at 12.9 ft. The shaft horse-power has increased, however, from 2582 to 2633 in passing from the lower to the higher projected area ratio, and this increase in projected area has only resulted in a decrease in efficiency of propulsion.

Therefore, the propeller to be used should be the 17-ft. diameter propeller having a projected ara ratio of .34 and the shaft and horse-power required will be approximately 2600.

Problem 27.—Incomplete Data

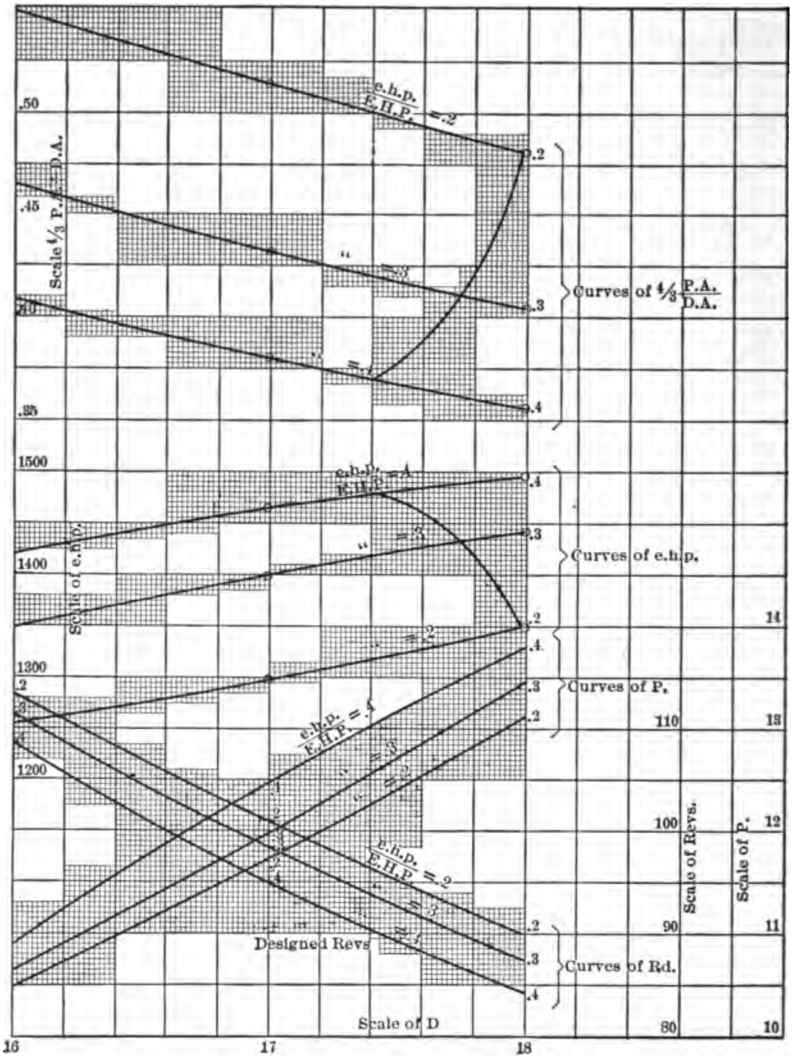
Same vessel as in Problem 23. Shaft horse-power of turbine reduction gear engine equals 2600. Expected speed 11 knots. Desired revolutions 90. Find propeller characterisitcs. Maximum diameter of propeller that can be carried = 18 ft.

SOLUTION

	Propeller 4 Bladed.								
D	18′	18′	18′	17′	17′	17′	16′	16′	16′
e.h.p. ÷ E.H.P.	.2	.3	.4	.2	.3	.4	.2	.3	.4
Z	.7279	.5445	.4144	.7279	.5445	.4144	.7279	.5445	.4144
S.H.P.$_d$	2600	2600	2600	2600	2600	2600	2600	2600	2600
K	1.27	1.27	1.27	1.27	1.27	1.27	1.27	1.27	1.27
S.H.P.$_p$	2047	2047	2047	2047	2047	2047	2047	2047	2047
S.H.P.	10941	7173	5316	10941	7173	5316	10941	7173	5316
I.H.P.	11893	7796	5778	11893	7796	5778	11893	7796	5778
v	11	11	11	11	11	11	11	11	11
$v \div V$ $\left(\text{Low}\frac{\text{e.t.}}{\text{E.T.}}\right)$	.652	.74	.805	.652	.74	.805	.652	.74	.805
V	16.87	14.87	13.67	16.87	14.87	13.67	16.87	14.87	13.67
I.T.$_D \div (1-S)$	5.42	4.031	3.25	6.076	4.519	3.643	6.86	5.102	4.113
P.A. ÷ D.A.	.36	.303	.267	.386	.324	.284	.412	.347	.306
$\frac{4}{3}$ P.A. ÷ D.A.	.48	.404	.356	.574	.432	.380	.548	.464	.408
P.C.	.567	.616	.647	.545	.597	.632	.527	.577	.614
E.H.P.	6743	4802	3738	6482	4654	3652	6268	4498	3548
e.h.p.	1349	1441	1495	1296	1396	1461	1254	1349	1419
T.S.	7970	6730	5930	8350	7150	6300	8820	7610	6800
$1-S$	.937	.942	.944	.933	.940	.943	.930	.930	.942
P	13′.11	13′.44	13′.99	11′.72	11′.97	12′.45	10′.48	10′.61	10.′87
A_V	3.68	3.51	3.4	3.68	3.51	3.4	3.68	3.51	3.4
A_v	3.12	3.12	3.12	3.12	3.12	3.12	3.12	3.12	3.12
s	.05436	.05161	.05219	.05781	.50339	.05312	.0604	.05517	.05405
R_d	89.92	87.45	84.04	101.0	98.35	94.53	113.3	111.2	108.4

Plotting these results upon values of D as abscissas, and running cross curves of P, $\frac{4}{3}$ P.A. ÷ D.A. and e.h.p. for $R = 90$ (Fig. 14), a series of propellers will be obtained of which the following are examples:

D	17′.4	17′.6	17′.8	17′.97
P	13′.07	13′.07	13′.07	13′.07
$\frac{4}{3}$ P.A. ÷ D.A.	.37	.39	.4225	.481
R	90	90	90	90
I.H.P.	2600	2600	2600	1600
e.h.p.	1475	1455	1413	1345
P.C.	.567	.56	.543	.517



Scale 4/3 P.A.÷D.A.
.50
.45
.40
.35
1500
1400
1300
1200
Scale of e.h.p.
e.h.p./E.H.P. = .2
= .3
= .4
Curves of 4/3 P.A./D.A.
Curves of e.h.p.
Curves of P.
Curves of Rd.
Designed Revs
Scale of D
Scale of Revs.
Scale of P.
110
100
90
80
14
13
12
11
10
16
17
18

FIG. 14.

Comparing the results obtained by this latter method with those obtained by the previous, it will be seen that as the effective horse-power, revolutions and speed of ship remain constant, the efficiency increases while the pitch and projected area ratios slowly decrease as the diameter of the propeller increases, while in the latter case, designing for constant engine or shaft power and constant revolutions and speed of ship, the efficiency falls, the pitch remains constant and the projected area ratio increases as the diameter increases.

The second case is a case of guesswork, pure and simple, depending entirely upon the accuracy of estimate of power required for speed for any given hull. Should this estimate be incorrect, the designer of the propeller would be made to shoulder the blame which should in reality rest upon other than his. In many cases care is taken to specify such an excess of engine power as to insure the desired speed of vessel thus unduly increasing the cost and weight of the machinery installation, an extravagance which could easily be avoided by the expenditure of a few hundred dollars for the construction of a model of the prospective vessel and trials of it in a model tank in order that a solid foundation upon which to design the necessary machinery be established.

Problem 28

Heavy vessel of intermediate speed. Showing effect of varying trim on *K*. Vessel fine lined at bow and stern, full midship section.

Characteristics of vessel—L.L.W.L. = 450′; *H* = 24′.5; *B* = 76.83; Midship Section Coef. = .196; Nominal B.C. = .66; *B* ÷ L.L.W.L = .171: Slip B.C. (Twin Screw) = .655; Prismatic Coef. = .787. This type of vessel was tried over the measured mile course with four different propellers and at four different times, as follows:

	1st Propeller.	2d Propeller.	3d Propeller.	4th Propeller.
D....................	17′.25	17′.25	17′.75	17′.33
P....................	18′	18′	18′	17.5
Blades................	3	3	3	3
P.A. ÷ D.A............	.308	.308	.315	.364

Displacements are all equal.
Base Line = Horizontal Line tangent to lowest point of keel at 24′.5.

Trim................	31½″ by stern	25½″ by stern	8″ by stern	2½″ by bow
Mean tip clearnace corrected for trim.......	3′.2	2′.85	2′.05	1′.65
K for M.T.C..........	1.08	1.10	1.19	1.31

These changes in the value of K appear abnormal and beyond the limits of possibility and therefore are apt to be charged up against other than the true cause, such as improper blade shape or blade section. If, however, either of these were the cause of the difference in propulsive efficiency, the revolutions obtained by using the actual power in the equations for apparent slip and estimate of revolutions, would differ widely from the actual revolutions as anything which changes the resistance of the blade to revolving, either increasing or decreasing it, would cause the estimated revolutions to vary widely from the actual ones.

ANALYSES OF PERFORMANCE OF ABOVE PROPELLERS

T.S.	6810	6810	6960	7750
$1-S$	.909	.909	.908	.902
V	20.29	20.29	20.13	22.18
I.T$_D$	3.87	3.87	4.07	5.15
I.H.P.	8926	8926	9873	132c5
P.C.	.677	.677	.673	.642
E.H.P.	6043	6043	6644	8477

PERFORMANCE AT CONTRACT SPEED OF 18 KNOTS

v	18	18	18	18
e.h.p.	4800	4800	4800	4800
e.h.p. ÷ E.H.P.	.7943	.7943	.7224	.5662
Z	.101	.101	.15	.261
I.H.P.$_v$	7064	7074	6989	7240
I.H.P.$_d$ Actual	7550	7875	8250	9500
I.H.P.$_d$ Est.	7640	7782	8317	9484
log A_V	3.91	3.91	3.90	4.015
log A_v	3.76	3.76	3.76	3.76
s	.11	.1121	.1070	.1266
R_d Actual	114.4	117.5	115.6	120.6
R_d Est.	113.9	114.1	113.4	119.3

As a further proof of what may be called the instability of the thrust deduction factor with this class of vessel, on the final acceptance trial of the vessel fitted with No. 4 propeller, the vessel was displacing 660 tons more than on the previous trial, yet the trial results obtained were:

I.H.P. per propeller	8075
Revolutions	118.85
Speed	17.81

The vessel was trimmed 2 ft. 5 in. by the stern, and these results check by analysis of power and revolutions as being produced by change in trim only.

This "instability in the value of K" existing, it becomes necessary to determine a standard condition of trim in order to design consistently. The condition of even trim fore-and-aft, is usually taken. The design conditions for the foregoing vessel were: Speed, 18 knots; revolutions 115 at 8250 I.H.P. and effective horse-power 4800 on each propeller. Center of propeller hub 10′ $7\frac{1}{2}$″ from the stern of the ship and 9′.5 above base. Propellers in Position 1.

DESIGN

D	17′	17′	17′	17′.5	17′.5	17′.5	18′	18′	18′
e.h.p. ÷ E.H.P.	.6	.7	.8	.6	.7	.8	.6	.7	.8
e.h.p.	4800	4800	4800	4800	4800	4800	4800	4800	4800
E.H.P.	8000	6857	6000	8000	6857	6000	8000	6857	6000
$v \div V$	.837	.877	.917	.837	.877	.917	.837	.877	.917
v	18	18	18	18	18	18	18	18	18
V	21.51	20.53	19.63	21.51	20.53	19.63	21.51	.2053	19.63
(P.A. ÷ D.A.) × E.T.$_p$	3.707	3.329	3.046	3.498	3.142	2.875	3.307	2.97	2.717
P.A. ÷ D.A.	.366	.338	.318	.351	.335	.305	.337	.312	.294
P.C.	.64	.659	.671	.65	.667	.678	.66	.675	.685
I.H.P.	12500	10405	8942	12307	10320	8850	12121	10159	8759
T.S.	8000	7450	7020	7700	7170	6770	7430	6910	6550
S.B.C.	.655	.655	.655	.655	.655	.655	.655	.655	.655
$1-S$	.902	.905	.908	.903	.906	.909	.905	.908	.910
P	16′.13	16′.48	17′.05	17′.25	17′.6	17′.77	18′.33	18′.74	18′.87
Hor. Tip Clear	...	2′.125	...	...	1.875	...	...	1.625	
Im. Up. Tips	...	6′.5	...	...	6′.25	...	...	6.0	
M.T.C.	...	2′.5	...	...	2′.32	...	...	1.96	
K	...	1.13	...	...	1.162	...	...	1.21	
Z	.231	.161	.1009	.231	.161	.1009	.231	.161	.1009
I.H.P.$_p$	7344	7182	7088	7231	7096	7015	7121	7012	6943
I.H.P.$_d$	8298	8116	8010	8402	8245	8151	8617	8484	8401
log A_V	3.98	3.91	3.87	3.98	3.91	3.87	3.98	3.91	3.87
log A_v	3.76	3.76	3.76	3.76	3.76	3.76	3.76	3.76	3.76
S	.10[illegible]	.1047	.1062	.1099	.1065	.108	.1121	.1085	.1112
R_d	126.8	123.7	119.7	118.7	116	115.1	112.1	109.2	108.8

Laying down the P, P.A. ÷ D.A., I.H.P.$_d$, and R_d on D as abscissas, and choosing $D = 17'.25$ as the diameter desired, arbitrarily choosing it as it was that of the best of the propellers in the preceding analysis, we find the following propeller:

$D = 17'.25$, $P = 17'.57$, P.A. ÷ D.A. = 308, I.H.P.$_d$ = 8105, $R_d = 116.4$, $V = 19.63$, e.h.p. ÷ E.H.P. = .8, $v \div V = .917$, e.t. ÷ E.T. = .872.

Maintaining constant $D = 17'.25$, P.A. ÷ D.A. = .308, $v = 18$, e.h.p = 4800, and e.t. ÷ E.T. = .872, we obtain

D	17.25	17.25	17.25	
v	18	18	18	
e.h.p.	4800	4800	4800	
e.h.p. ÷ E.H.P.	.75	.70	.65	
Z	.130	.161	.2	
E.H.P.	6400	6857	7385	
P.A. ÷ D.A.	.308	.308	.308	
P.C.	.677	.677	.677	
I.H.P.	9454	10128	10908	
I.H.P.$_p$	7008	6991	6883	
K	1.14	1.14	1.14	
I.H.P.$_d$	7989	7970	7846	8105
$v \div V$	.86	.803	.747	
V	20.93	22.42	24.10	
S.B.C.	.655	.655	.655	
$1-S$	.91	.91	.91	
T.S. for $\frac{\text{P.A.}}{\text{D.A.}}$	6850	6850	6850	
P	18'.44	19.75	21'.23	
$\log A_v$	3.94	4.03	4.1	
$\log A_s$	3.76	3.76	3.76	
s	.1151	.1319	.1416	
R_d	111.8	106.4	100.1	116.4*
I.H.P.$_{d_1}$	8250	8250	8250	8250*
	(Designed Power of Engine)			
$R_{d_1} = R_d \sqrt[3]{\frac{\text{I.H.P.}_{d_1}}{\text{I.H.P.}_d}}$	113	107.6	101.8	117.1*

* From preceding calculation.

Plotting these results on P as abscissas, the following propeller is obtained as filling the required conditions:

$$D = 17'.25 \qquad R = 115$$
$$P = 18' \qquad \text{I.H.P.}_{d_1} = 8250$$
$$\text{P.A.} \div \text{D.A.} = .308$$

while at 18 knots, the results will be $R_d = 114$, I.H.P.$_d = 8025$, the differences between revolutions and power between these results and those of the initial analysis being caused by difference in the values of K used, and very slight difference in the value of $1-S$.

Problem 29

The vessel of Problem 28 was of a type of hull whose afterbody fines rapidly both from the keel up and from the beam in towards the center line, the midship section being very full. The vessel in Problem 29 is, however, of an entirely different type, the stern being of the type commonly called "fan-tail," the diminution of beam at upper deck at the propellers being comparatively small, while the fining of the afterbody lines occurs chiefly in a rapid rise from the flat bottom of the middle body. The hull of the ship above the propellers is well above the water plane. The propellers are, therefore, in Position 2.

The dimensions of the vessel are

L.L.W.L. = 520′	Block Coef. = .7215
B = 65′ 02 $\frac{11}{16}$″	Coef. Mid. Sec. = .9783
H = 27′ 6″	Coef. L.W. Plane = .7954
Displace. = 19230 tons	Cyl. Coef. = .7375

Twin Screw

B ÷ L.L.W.L. = .126

Slip B.C. = .617

Des. Speed = 14 knots.

Max. Diam. of Propeller = 16′, Revs. at designed speed = 108, e.h.p. for designed speed = 2550 on two screws.

For such a heavy vessel and on account of the character of her service, which is collier, as it is desirable to carry as great a diameter of propeller as possible in order to obtain the maximum handling, backing and holding power in bad weather, we, therefore, select for the first approximation, the upper design curve shown on Sheet 22, using the maximum diameter, 16′, that can be carried.

COMPUTATION

e.h.p. ÷ E.H.P.	.3	.4	.5	.6	.7
e.h.p.	1775	1775	1775	1775	1775
E.H.P.	5917	4438	3550	2960	2536
v	14	14	14	14	.933
$v \div V$	.742	.807	.86	.901	14
V	18.87	17.35	16.28	15.54	15.01
D	16′	16′	16′	16′	16′
Blades	3	3	3	3	3
(P.A. ÷ D.A.) × E.T.$_p$ (Sheet 24)	3.464	2.819	2.411	2.104	1.867
P.A. ÷ D.A.	.347	.30	.269	.2475	.23
P.C.	.654	.682	.695	.702	.706
I.H.P.	9047	6508	5108	4315	3592
S.H.P.	8324	5987	4699	3970	3305
T.S for P.A. ÷ D.A.	7600	6660	5950	5430	5000
S.B.C.	.617	.617	.617	.617	.617
$1-S$	.899	.903	.905	.906	.906
P	14.07	14.69	15.4	16.09	16.87
Z for $\frac{\text{e.h.p.}}{\text{E.H.P.}}$	.5445	.4144	.3135	.231	.161
I.H.P.$_p$	2582	2506	2482	2535	2480
S.H.P.$_p$	2376	2306	2283	2332	2281
K from $C-C_{-}$ (Sheet 19)	1.165	1.165	1.165	1.165	1.165
I.H.P.$_d$	3009	2920	2891	2953	2889
S.H.P.$_d$	2768	2686	2660	2717	2658
Log A_V	3.82	3.71	3.63	3.57	3.52
Log A_v	3.43	3.43	3.43	3.43	3.43
s	.08258	.08291	.08522	.08881	.093
R_d	109.9	105.3	100.7	96.78	92.71

The actual propeller fitted to this vessel and its performances, was as follows:

Propeller	
D	15'.95
P	14'.436
Blades	3
S.H.P_d	2730
Revs	108.5
P.A.÷D.A.	.304

Laying down $v \div V$ on P.A.÷D.A. as abscissas from the above calculations, in order to check the accuracy of the design charts by comparing the computed performance of propellers having the same pitch and projected area ratio as the actual propellers, take off the value of $v \div V$ corresponding to P.A.÷D.A. equal .304, which is found to be .802, while the value of e.h.p.÷E.H.P., also laid down as a curve, is found to be .39

The line drawn through this point plotted on Sheet 22, and zero, corresponds to a value e.t.÷E.T. =.49.

D	16'	16'	16'
e.t.÷E.T.	.49	.49	.49
e.h.p.÷E.H.P.	.4	.425	.45
$v \div V$	.82	.87	.92
e.h.p.	1775	1775	1775
E.H.P.	4438	4177	3945
P.A.÷D.A.	.304	.304	.304
P.C.	.679	.679	.679
I.H.P.	6535	6151	5809
S.H.P.	6013	5659	5345
Z	.4144	.387	.362
S.H.P_p	2316	2321	2322
K	1.165	1.165	1.165
S.H.$P._d$	2698	2704	2706
T.S.	6750	6750	6750
$1-S$	.902	.902	.902
v	14	14	14
V	17.07	16.09	15.22
P	14'.28	13'.46	12'.73
log A_V	3.7	3.61	3.54
log A_v	3.43	3.43	3.43
s	.08188	.07094	.06391
R_d	108.2	113.4	119.1

Laying down curves of P, S.H.$P._d$, and R_d on P.A.÷D.A. as abscissas the propeller for 108.5 revolutions is found to be, from the first set of calculations:

$D = 16'$, $P = 15' 9''$, P.A.÷D.A. = .257, S.H.$P._d$ on each propeller = 2700, $R_d = 108.5$, $v = 14$ knots.

Laying down curves of P, R_d and S.H.P.$_d$ on P as abscissas, constant value of P.A. ÷ D.A. = .304, from the second set of calculations, the resultant propeller for 108.5 revolutions is found to be:

$D=16'$, $P=14'\ 3''$, P.A. ÷ D.A. = .304, S.H.P.$_d$ = 2700 on each propeller, $R_d=108.5$, $v=14$ knots.

This latter propeller is found to agree very closely in all particulars and in promised performance, with the actual propeller, while by comparing it with the propeller obtained by the first calculation, pitch is seen to have been exchanged for surface while the efficiency has remained constant. Surface being desirable for manœuvring power, it would be desirable to choose the second propeller rather than the first. The weights of the two propellers would probably be in favor of the higher pitch propeller.

Problem 30

Destroyer; Slip B.C. = .385; twin screw; designed speed = 37 knots; e.h.p. (total) of hull and appendages, for this speed = 19250; designed revolutions not less than 495; find S.H.P.$_d$, P, and P.A. ÷ D.A., the diameter that can be carried being 113 inches.

In problems of high power and speed, in order to hold the propeller within proper and practical limits of projected area ratio and diameter it becomes necessary to design at or near the natural speed and load limits, the term natural here used meaning the limits imposed by Sheet 22 where the cavitation condition is that imposed by the e.t. line e.t. = 1.225 E.T., and not the curve E.T.

For primary calculation use the curve on Sheet 22, marked "Safe Limit for High Efficiency."

e.h.p. ÷ E.H.P.	.9	.95	1.0	1.05	1.10
e.h.p.	9625	9625	9625	9625	9625
E.H.P	10694	10132	9625	9167	8750
v	37	37	37	37	37
$v \div V$	.952	.967	.981	.997	1.01
V	38.87	38.26	37.72	37.11	36.63
$D=113''$	9'.42	9'.42	9'.42	9'.42	9'.42
(P.A. ÷ D.A.) × E.T.$_p$ (3 blades)	8.929	8.283	8.283	8.017	7.752
P.A. ÷ D.A	.637	.624	.612	.601	.591
P.C.	.525	.525	.525	.525	.525
I.H.P	20370	19298	18333	17460	16667
S.H.P	18741	17755	16867	16064	15334
T.S.	15610	14550	14000	13560	13200
$1-S$	.812	.822	.828	.835	.841
P	9'.195	9'.594	9'.757	9'.829	9'.896
Z	−.0477	−.0224	0	+.0221	+.0431
K	1	1	1	1	1
S.H.P.$_d$ = S.H.P.$_p$	16792	16862	16867	16902	16933

In estimating the revolutions for this type of vessel, where under high speed there is a liability to excessive squatting of the stern, attention must be paid to the construction of the horizontal arms of the propeller struts. Where the long axis of the sections of these arms are parallel to the base line of the vessel, that is, horizontal at normal trim, the tendency to squat is much reduced and the wake conditions tend to remain normal. In such cases the values of Log A_V and of Log A_v are both taken from the normal curve X on Sheet 21.

Should the axes of the sections of the horizontal strut arms be inclined downward at the forward ends in order to get them into the lines of flow, the squatting of the stern is augmented, the wake rapidly decreases as the speed increases and the revolutions increase rapidly. The augmentation of revolutions begins when $v \div \sqrt{\text{L.L.W.L.}} = 1.48$ and at this point the values of Log A_v begin to depart from the curve X, Sheet 21, moving towards the curve Y which they reach when $v \div \sqrt{\text{L.L.W.L.}} = 1.75$. The value of Log A_V is in all cases taken from the curve X. This same phenomenon will occur where the strut arm sections are parallel to the base should the propellers be located as far aft as the stern post.

In the case in question, let us estimate the revolutions for both conditions, 1st, limited squat; 2d, excessive squat.

Log A_V (X, Sheet 21)	4.48	4.47	4.465	4.46	4.450
Log A_v X, Sheet 21	4.455	4.455	4.455	4.455	4.455
Log A_v Y, Sheet 21	4.33	4.33	4.33	4.33	4.33
s 1st	.1784	.175	.176	.1756	.1756
s 2d	.2656	.2457	.2347	.2342	.2315
R_d 1st	496.3	473.7	466.3	462.7	459.6
R_d 2d	555.2	518.1	502.1	498.1	493

The propellers to give 495 revolutions under the designed conditions of speed are

	A For Squat.	B For no Squat.
D	113″	113″
P	118″.5	110″.5
P.A.÷D.A	.595	.6365
Blades	3	3
S.H.P.$_d$	16920	16800
R_d	495	495
v	37	37

Now, let us suppose that the propellers are placed as far forward on the afterbody as possible and still retain large tip clearance, and that the axes

of the lower strut arms are parallel to the base line of the vessel. The tendency to squat will be much reduced, the increase in immersion of the propellers due to squatting which will occur to some extent will be small and the propellers will be in the best position for realizing the maximum benefit of the wake.

By doing this, the propeller A is eliminated and propeller B may be chosen, although not necessarily, as we may extend our choice, as follows:

Take e.t. ÷ E.T. for $\frac{\text{e.h.p.}}{\text{E.H.P.}}$ and $\frac{v}{V}$ (Sheet 22), corresponding to above computations. Draw lines through zero and these points on Sheet 22, and with constant P.A. ÷ D.A. and constant D and varying $v \div V$ and e.h.p. ÷ E.H.P., taken from each of these lines of e.t. ÷ E.T., obtain a series of propellers for each of the values of P.A. ÷ D.A. obtained in the first calculations, and obtain from each series that propeller giving 495 revolutions. This will allow a cross curve of propellers of constant diameter but of varying P.A. ÷ D.A. varying pitch and varying S.H.P.$_d$, but all of constant revolutions, 495, for constant speed, 37 knots, from which we may make our choice.

SOLUTION

$\frac{\text{e.h.p}}{\text{E.H.P}}$	1.0	1.05	1.1	1.15	1.15	1.20
$v \div V$	1.02	1.075	1.05	1.1	1.06	1.11
P.A. ÷ D.A	.624	.624	.601	.601	.591	.591
e.h.p.	9625	9625	9625	9625	9625	9625
E.H.P	9625	9167	8750	8370	8370	8021
P.C.	.525	.525	.525	.525	.525	.525
I.H.P	18333	17460	16667	15943	15943	15728
S.H.P	16867	16064	15334	14668	14668	14470
Z	0.00	+.0221	+.0431	+.0632	+.0632	+.0825
S.H.P.$_d$	16867	16902	16934	16965	16965	17497
v	37	37	37	37	37	37
V	36.28	34.42	35.24	33.64	34.91	33.34
T.S.	14550	14550	13560	13560	13200	13200
$1-S$	.822	.822	.835	.835	.841	.841
P	9′.095	8′.63	9′.333	8′.909	9.429	9.004
Log A_V	4.445	4.415	4.43	4.395	4.42	4.39
Log A_v	4.455	4.455	4.455	4.455	4.455	4.455
s	.174	.1708	.1720	.1662	.1697	.1655
R_d	499.1	523.9	485.2	504.8	478.9	499

Laying down these results on values of $\frac{\text{e.h.p.}}{\text{E.H.P.}}$ as abscissas, a series of curves P, P.A. ÷ D.A. and of S.H.P.$_d$ are obtained for the constant diameter

113 in., constant revolutions, 495, and constant e.h.p., 9625 on each propeller, from which the following table of propellers may be prepared:

D	P.A. ÷ D.A.	P	R_d	Total S.H.P.$_d$	e.h.p.	p.c.	v
113″	.59	110″	495	34700	19250	.5547	37
113″	.595	110″½	495	34050	19250	.5653	37
113″	.600	111″	495	33900	19250	.5678	37
113″	.605	111″⅜	495	33880	19250	.5682	37
113″	.610	111″⅝	495	33850	19250	.5687	37
113″	.615	112″	495	33800	19250	.5695	37
113″	.620	112″⅜	495	33780	19250	.5698	37
113″	.625	112″⅞	495	33760	19250	.5700	37

In selecting the propeller to use it will be advisable to take one of the heavier projected area ratios as they not only promise slightly higher propulsive efficiencies but have the added advantage of greater range before cavitation is encountered.

Should squatting occur to any great extent, the revolutions will speed up until a sufficient degree of squat has been obtained to shift the factor Log A_v from the X to the Y curve, Sheet 21, when no further increase will occur. To cover this contingency it may be considered desirable to design the propeller for conditions of wake at the designed speed corresponding to a position intermediate to curve X and Y so that the decrease or increase in revolutions will not be excessive.

Design of Propellers having blades not of Standard Form.

It has already been pointed out how such propellers may be divided into three cases for analysis. The same distinction can be made as to design and the forms for computation modified accordingly.

In the first forms the resultant propeller is designed to deliver the same effective (tow-rope) horse-power as the basic propeller of diameter D_1, but does so at an increase in power and revolutions over those of the basic. In the second form, the effective horse-power delivered varies directly as the $\frac{3}{2}$ power of the ratio between the actual and basic diameters, and the powers vary according to the square of the same ratio, the revolutions increase inversely as the $\frac{1}{4}$ power of the diameter ratios. The propulsive coefficient of the actual propeller will be to that of the basic propeller as the square root of the inverse ratio of the diameters.

FAN-SHAPED BLADES: FORMS FOR CASE 1: REVOLUTIONS LESS THAN STANDARD FOR POWER, DIAMETER AND SPEED

Condition	Constant I.H.P.$_d$			Constant e.h.p.		
Diameter possible	D (constant).			D (constant)		
Diameter (Basic Prop., assumed)	D_1			D_1		
Basic Power	I.H.P.$_{d_1}$ = I.H.P.$_d \times \frac{D}{D_1}$			e.h.p.$_1$ = e.h.p.		
e.h.p.$_1$ ÷ E.H.P.$_1$	( )	( )	( )	( )	( )	( )
E.H.P.$_1$	....			( )	( )	( )
K	Constant			Constant		
I.H.P.$_{p_1}$ = I.H.P.$_{d_1} \div K$	Constant					
Z for e.h.p.$_1$ ÷ E.H.P.$_1$	( )	( )	( )	( )	( )	( )
I.H.P.$_1$ = I.H.P.$_{p_1} \times 10^Z$	( )	( )	( )			
Designed speed	v = Constant			v = Constant		
$v \div V$	( )	( )	( )	( )	( )	( )
V	( )	( )	( )	( )	( )	( )
I.T.$_D \div (1-S)$	$(C \times$ I.H.P.$_1) \div (D_1^2 \times V)$					
(P.A. ÷ DA.) × E.T.$_p$	....			$(C \times$ E.H.P.$_1) \div (D_1^2 \times V)$		
Note: $C = \begin{cases} 3.84, \text{ 2 Blades} \\ 2.88, \text{ 3 Blades} \\ 2.491, \text{ 4 Blades} \end{cases}$						
P.A. ÷ D.A. (Sheets 23 and 24)	( )	( )	( )	( )	( )	( )
$\frac{2}{3}$ P.A. ÷ D.A. (2 Blades)	( )	( )	( )	( )	( )	( )
$\frac{4}{3}$ P.A. ÷ D.A. (4 Blades)	( )	( )	( )	( )	( )	( )
P.C. for total proj. area ratio	( )	( )	( )	( )	( )	( )
E.H.P. = I.H.P.$_1$ × P.C.	( )	( )	( )			
e.h.p.$_1$ = E.H.P.$_1 \times \left(\frac{\text{e.h.p.}}{\text{E.H.P.}_1}\right)$	( )	( )	( )			
e.h.p. = e.h.p.$_1$	( )	( )	( )			
I.H.P. = E.H.P.$_1$ ÷ P.C.	....			( )	( )	( )
I.H.P.$_{p_1}$ = I.H.P.$_1 \div 10^Z$	....			( )	( )	( )
I.H.P.$_{d_1}$ = I.H.P.$_{p_1} \times K$	....			( )	( )	( )
I.H.P.$_d$ = I.H.P.$_{d_1} \times \frac{D_1}{D}$	....			( )	( )	( )
T.S. for P.A. ÷ D.A.	( )	( )	( )	( )	( )	( )
S.B.C. of vessel	Constant			Constant		
$1-S$ for $\frac{\text{P.A.}}{\text{D.A.}}$ and S.B.C.	( )	( )	( )	( )	( )	( )
$P = \frac{101.33 \times V \times \pi \times D}{\text{T.S.} \times (1-S)}$	( )	( )	( )	( )	( )	( )
Log. A_V for V (Sheet 21)	( )	( )	( )	( )	( )	( )
Log. A_v for v (Sheet 21)	( )	( )	( )	( )	( )	( )
$s_1 = S \frac{\text{I.H.P.}_{d_1} \times A_V}{\text{I.H.P.}_1 \times A_v}$	( )	( )	( )	( )	( )	( )
$R_{d_1} = \frac{v \times 101.33}{P \times (1-s_1)}$	( )	( )	( )	( )	( )	( )
R_d = Revs. of Actual Screw	$= R_{d_1} \times \left(\frac{D_1}{D}\right)^{1/2}$			$R_{d_1} \times \left(\frac{D_1}{D}\right)^{1/2}$		
Proj. Area Ratio (Actual) = Total Proj. Area Ratio of Basic Propeller × $(D_1 \div D)^2$						

OVAL BLADES BROADER OR NARROWER AT TIPS THAN STANDARD. FORMS FOR CASES 2 AND 3. REVOLUTIONS GREATER OR LESS THAN STANDARD FOR POWER, DIAMETER, AND SPEED

Condition	Constant I.H.P.$_d$	Constant e.h.p.
Diameter possible	D (constant)	D (constant)
Diameter (Basic, assumed)	D_1	D_1
Basic Power	I.H.P.$_{d_1}$ = I.H.P.$_d$ $\times (D_1 \div D)^2$	e.h.p.$_1$ = e.h.p. $\times (D_1 \div D)^{3/2}$
e.h.p.$_1$ ÷ E.H.P.$_1$	( ) ( ) ( )	( ) ( ) ( )
E.H.P.$_1$	....	( ) ( ) ( )
K	Constant	Constant
I.H.P.$_{p_1}$ = I.H.P.$_{d_1}$ ÷ K	Constant	
Z for e.h.p.$_1$ ÷ E.H.P.$_1$	( ) ( ) ( )	( ) ( ) ( )
I.H.P.$_1$ = I.H.P.$_{p_1}$ $\times 10^Z$	( ) ( ) ( )	
Designed Speed = v	v = Constant	v = Constant
$v \div V$	( ) ( ) ( )	( ) ( ) ( )
V	( ) ( ) ( )	( ) ( ) ( )
I.T.$_D$ ÷ $(1-S)$	(C×I.H.P.$_1$) ÷ $(D_1^2 \times V)$	
(P.A. ÷ D.A.) × E.T.$_p$	....	(C×E.H.P.$_1$) ÷ $(D_1^2 \times V)$
Note: C as before.		
P.A. ÷ D.A. (Sheets 23 and 24)	( ) ( ) ( )	( ) ( ) ( )
$\frac{2}{3}$ P.A. ÷ D.A. for 2 Blades	( ) ( ) ( )	( ) ( ) ( )
$\frac{4}{3}$ P.A. ÷ D.A. for 4 Blades	( ) ( ) ( )	( ) ( ) ( )
P.C. for Total Proj. Area Ratio	( ) ( ) ( )	( ) ( ) ( )
E.H.P.$_1$ = I.H.P.$_1$ × P.C	( ) ( ) ( )	
I.H.P.$_1$ = E.H.P.$_1$ ÷ P.C	....	( ) ( ) ( )
I.H.P.$_{p_1}$ = I.H.P.$_1$ $\div 10^Z$	....	( ) ( ) ( )
I.H.P.$_{d_1}$ = I.H.P.$_{p_1}$ × K	....	( ) ( ) ( )
I.H.P.$_d$ = I.H.P.$_{d_1}$ $\times \left(\frac{D}{D_1}\right)^2$	....	( ) ( ) ( )
e.h.p.$_1$ = E.H.P.$_1$ × (e.h.p.$_1$ ÷ E.H.P.$_1$)	( ) ( ) ( )	
e.h.p. = e.h.p.$_1$ $\times (D \div D_1)^{3/2}$	( ) ( ) ( )	
T.S. for P.A. ÷ D.A.	( ) ( ) ( )	( ) ( ) ( )
S.B.C. of Vessel	Constant	Constant
$1-S$ for $\frac{\text{P.A.}}{\text{D.A.}}$ and S.B.C	( ) ( ) ( )	( ) ( ) ( )
$P = \frac{101.33 \times V \times \pi \times D_1}{\text{T.S.} \times (1-S)}$	( ) ( ) ( )	( ) ( ) ( )
Log A_V	( ) ( ) ( )	( ) ( ) ( )
Log A_v	( ) ( ) ( )	( ) ( ) ( )
$s = S \frac{\text{I.H.P.}_{d_1} \times A_V}{\text{I.H.P.}_1 \times A_v}$	( ) ( ) ( )	( ) ( ) ( )
$R_{d_1} = \frac{101.33 \times v}{P \times (1-s)}$	( ) ( ) ( )	( ) ( ) ( )
$R_d = R_{d_1} \times \left(\frac{D_1}{D}\right)^{1/4}$	( ) ( ) ( )	( ) ( ) ( )

The limit that can be put to this change in diameter is not known, but it is believed that a difference not exceeding from 15 to 20 per cent between the actual and the basic diameters can be used without any material error being introduced.

Problem 31

In order to illustrate the foregoing methods of design, let the case of a tow boat be taken for which the data are as follows:

Hull Conditions
Displ. = 1147 tons
L.L.W.L. = 185′ 2″
$B = 34' \ 1\frac{1}{2}''$
$H = 12' \ 6''$
$B \div$ L.L.W.L. = .184
Nominal B.C. = .531
Coef. Mid. Sec. = 875
Slip B.C. (single screw) = .755
K (curve $C-C-C_2$, Sheet 21) = 1.28
Propeller to be four-bladed

Power and Propeller
Designed I.H.P. = 1800
Designed Revs. = 120
Max. Diam. of Prop. = 12′
Expected sea speed = 14 knots
To tow efficiently at 10 knots, engine assumed to be able to develop full power at this speed.
e.h.p. for 14 knots = 1130

In solving this problem, solve for both Case 1 and Case 2, using designed full power of the engine and then solve in Problem 32 again for both cases of diameter reduction, using the e.h.p. for 14 knots.

FULL POWER OF ENGINE

Case.	1			2		
I.H.P.$_d$	1800	1800	1800	1800	1800	1800
D	12′	12′	12′	12′	12′	12′
D_1	12′	13′	14′	12′	13′	14′
$D_1 \div D$	1.0	1.0833	1.1667	1.0	1.0833	1.1667
$(D_1 \div D)^{1/2}$	1.0	1.041	1.080	1.0		
$(D_1 \div D)^2$	1.0	....	....	1.0	1.174	1.361
$(D_1 \div D)^{3/2}$	1.0	....	....	1.0	1.141	1.26
$(D_1 \div D)^{1/4}$	....	....	....	1.0	1.02	1.033
I.H.P.$_{d_1}$ = I.H.P.$_d \div \dfrac{D}{D_1}$	1800	1662	1543			
I.H.P.$_{d_1}$ = I.H.P.$_d \times \left(\dfrac{D_1}{D}\right)^2$	....	....	....	1800	2112	2450
K	1.28	1.28	1.28	1.28	1.28	1.28
I.H.P.$_{p_1}$ = I.H.P.$_{d_1} \div K$	1406	1298	1205	1406	1650	1914

In selecting the value of e.h.p. ÷ E.H.P. to use, as the vessel is required to tow efficiently at 10 knots when developing full power of the engine, the maximum value of e.h.p. ÷ E.H.P. for the slip block coefficient of the vessel and for 10 knots, obtained from Sheet 22B, should be used. This is seen to be, by interpolation, .3. ∴

e.h.p.$_1$ ÷ E.H.P.$_1$	.3	.3	.3	.3	.3	.3
Z	.5445	.5445	.5445	.5445	.5445	.5445
I.H.P.$_1$	4927	4548	4223	4927	5782	6706

To find the value of $v_1 \div V$ from which to obtain the value of V, v_1 being the towing speed, proceed:

Z^1 = Log I.H.P.$_1$ − Log I.H.P.$_{d1}$	.43729	.43729	.43729	.43729	.43729	.43729
e.h.p.^{1} ÷ E.H.P.$_1$ for Z^1	.382	.382	.382	.382	.382	.382

$v_1 \div V$ for $\frac{\text{e.h.p.}_1}{\text{E.H.P.}_1} = .382$ and $\frac{\text{e.t.}}{E.T.} = 1.0$ is only .283. ∴ no danger of cavitation. To avoid dispersion of thrust column for $\frac{\text{e.h.p.}}{\text{E.H.P.}} = .3$, take a value $\frac{v_1}{V}$ well above critical thrusts, say = .6

v_1	10	10	10	10	10	10
$v_1 \div V$	.6	.6	.6	.6	.6	.6
V	16.67	16.67	16.67	16.67	16.67	16.67
$\frac{\text{I.T.}_D}{1-S} = \frac{2.491 \text{ I.H.P.}_1}{D_1^2 \times V}$	5.113	4.021	3.219	5.113	5.113	5.113
P.A. ÷ D.A. for $\frac{\text{I.T.}_D}{1-S}$	.346	.30	.26	.346	.346	.346
$\frac{4}{3}$ P.A. ÷ D.A.	.460	.40	.347	.460	.460	.460
P.C. for $\frac{4}{3}\frac{\text{P.A.}}{\text{D.A.}}$	.58	.62	.654	.58	.58	.58
E.H.P.$_1$	2858	2820	2762	2858	3354	3890
e.h.p.$_1$	857	846	829	857	1006	1167
e.h.p. = e.h.p.$_1 \times \left(\frac{D}{D_1}\right)^{3/2}$	....	....	....	857	882	926
e.h.p. = e.h.p$_1$	857	846	829			
v for e.h.p.*	13.22	13.2	13.15	13.22	13.3	13.43

* Taken from model tank e.h.p. curve.

Vessel, therefore, can not make 14 knots with 1800 I.H.P.$_d$. In fact, the two sister ships built on these lines made 13.32 and 13.54 knots, respectively, with this power.

T.S. for $\frac{\text{P.A.}}{\text{D.A.}}$	7580	6650	5740	7580	7580	7580
$1-S$ for S.B.C. and $\frac{\text{P.A.}}{\text{D.A.}}$	.925	.93	.935	.925	.925	.925
$P=\frac{V\times 101.33\times\pi D}{\text{T.S.}\times(1-S)}$	9'.081	11'.14	13'.84	9'.081	9'.838	10'.6
Log A_V (Curve X, Sheet 21)	3.735	3.735	3.735	3.735	3.735	3.735
Log A_v (Curve Y, Sheet 21)	3.195	3.198	3.185	3.195	3.2	3.21
s_1	.09501	.0897	.08426	.09501	.09611	.09722
R_{d_1}	163	131.8	105.1	163	151.6	142.3
$R_d=R_{d_1}\times\left(\frac{D_1}{D}\right)^{1/2}$	163	137.1	113.7			
$R_d=R_{d_1}\times\left(\frac{D_1}{D}\right)^{1/4}$	....	....	....	163	154.6	147.9

By inspection it is at once seen that the propeller designed under Case 2 will not do, as their revolutions are all too high. From those designed by Case 1 a propeller giving the desired revolutions can be obtained, therefore, laying down these Case 1 propellers on D_1 values as abscissas, we obtain:

Propeller.	Basic.	Actual.
D	13'.725	12'
P	13'.06	13'.06
P.A.÷D.A.	.27	.3532
$\frac{4}{3}$ P.A.÷D.A.	.36	.4709 (Fan Shaped)
I.H.P.$_d$		1800
e.h.p.		834
P.C.		.463
v		13.16
R_d		120

To find the revolutions at 10 knots, supposing the engine to be able to develop its full power at this speed of vessel, proceed as follows:

Log A_V (Curve X, Sheet 21)	3.735	3.735	3.735
Log A_v (Curve Y, Sheet 21)	2.845	2.845	2.845
s_1	.2127	.1986	.1844
R_{d_1}	141.7	113.4	89.77
$R_d=R_{d_1}\times\left(\frac{D_1}{D}\right)^{1/2}$	141.7	113.1	96.95

Laying down these values of R_d on D_1 as abscissas it will be found that the revolutions of the propeller selected will be 102.4 at 10 knots speed and with the engine developing 1800 I.H.P.$_d$.

Comparing the two methods of diameter reduction it will be seen that for constant values of e.h.p. ÷ E.H.P. and of $v \div V$ and constant power, as the diameter reduction is increased,

	Case 1	*Case 2*
Proj. Area Ratio	Decreases	Constant
Revolutions	Decrease rapidly	Decrease slowly
Pitch	Increases rapidly	Increases slowly
Propulsive efficiency	Decreases slowly	Increases slowly

Where the desired revolutions are much below those which would be obtained with the desired diameter and power without diameter reduction, Case 1 is to be preferred. Where the reduction of revolutions and diameter are small Case 2 should be always used.

Problem 32

Same hull and speed requirements as in Problem 31. The effective horse-power for 14 knots equals 1130, as before. To obtain propeller characteristics, revolutions and I.H.P.$_d$ necessary for a sea speed, light, of 14 knots; and revolutions with this power for a towing speed of 10 knots.

Shape of Blade.	Case 1, Fan.			Case 2, Broad Tipped.		
D	12′	12′	12′	12′	12′	12′
D_1	12′	13′	14′	12′	13′	14′
e.h.p.	1130	1130	1130	1130	1130	1130
$D_1 \div D$	1.0	1.0833	1.1667	1.0	1.0833	1.1667
$(D_1 \div D)^{1/2}$	1.0	1.041	1.08			
$(D_1 \div D)^2$	....	....	....	1.0	1.174	1.361
$(D_1 \div D)^{3/2}$	....	....	....	1.0	1.141	1.26
$(D_1 \div D)^{1/4}$	....	....	....	1.0	1.02	1.033
e.h.p.$_1$ = e.h.p.	1130	1130	1130			
e.h.p.$_1$ = e.h.p. $\times \left(\frac{D_1}{D}\right)^{3/2}$	....	....	....	1130	1289.33	1423.8
e.h.p.$_1$ ÷ E.H.P. (as before)	.3	.3	.3	.3	.3	.3
E.H.P.$_1$	3767	3767	3767	3767	4298	4746
v_1 (Towing)	10	10	10	10	10	10
$v_1 \div V$ (as before)	.6	.6	.6	.6	.6	.6
V	16.67	16.67	16.67	16.67	16.67	16.67
(P.A. ÷ D.A.) × E.T.$_p$	3.91	3.332	2.873	3.91	3.801	3.619
P.A. ÷ D.A.	.388	.344	.31	.388	.38	.366
$\frac{4}{3}$ P.A. ÷ D.A.	.517	.457	.413	.517	.505	.488
P.C.	.544	.582	.61	.544	.551	.561
I.H.P.$_1$ = E.H.P.$_1$ ÷ P.C.	6925	6473	6176	6925	7801	8460
Z	.5445	.5445	.5445	.5445	.5445	.5445
I.H.P.$_{p_1}$	1977	1848	1763	1977	2227	2415
K	1.28	1.28	1.28	1.28	1.28	1.28
I.H.P.$_{d_1}$	2530	2365	2256	2530	2850	3091
I.H.P.$_d$ = I.H.P.$_{d_1} \times \left(\frac{D_1}{D}\right)$	2530	2562	2632			
I.H.P.$_d$ = I.H.P.$_{d_1} \div \left(\frac{D_1}{D}\right)^2$	....	....	....	2530	2428	2271
T.S.	8370	7520	6850	8370	8220	8000
S.B.C.	.755	.755	.755	.755	.755	.755
$1 - S$	.92	.926	.93	.92	.922	.923
P	8′.268	9′.905	11′.66	8′.268	9′.101	10′.06
v	14	14	14	14	14	14
Log A_V (Curve X, Sheet 21)	3.735	3.735	3.735	3.735	3.735	3.735
Log A_v (Curve Y, Sheet 21)	3.26	3.26	3.26	3.26	3.26	3.26
s_1	.08726	.08071	.07635	.08726	.08508	.08399
R_{d_1}	188	155.8	131.8	188	170.4	154
$R_d = R_{d_1} \times \left(\frac{D_1}{D}\right)^{1/2}$	188	162.2	142.3			
$R_d = R_{d_1} \times \left(\frac{D_1}{D}\right)^{1/4}$	....	....	....	188	173.8	159

In order to arrive at 120 revolutions a still greater diameter reduction would be required, therefore, in order to shorten the work, let us suppose the desired revolutions under the 14-knot condition are 160, then laying down both Case 1 and Case 2 on D_1 as abscissas we obtain the following propellers:

Condition.	Case 1		Case 2	
	Basic.	Actual.	Basic.	Actual.
Diameter	13'.1	12'	13'.925	12'
Pitch	10'.06	10'.06	9'.99	9'.99
P.A.÷D.A	.34	....	.3675	....
$\frac{4}{3}$ P.A.÷.D.A	.452	.5387	.490	....
Blades	4	4	4	4
$I.H.P._{d_1}$	2352	....	3780	....
$I.H.P._d$	....	2567	....	2288
R_{d_1}	153	....	155	....
R_d	....	160	....	160
$e.h.p._1$	1130	....	1415	....
$e.h._D$	....	1130	....	1130
$P.C. = \frac{e.h.p.}{I.H.P_d}$	....	.44	....	.494
v	14	14	14	14

The revolutions for 10 knots with these same values of $I.H.P._d$ can now be calculated, using Log A_0 from Curve Y, Sheet 21, and using the values of $I.H.P._1$, $I.H.P._{d_1}$, as obtained in the foregoing calculations.

It will be noted that where the desired revolutions can be obtained by the use of Case 2, without excessive reduction in diameter, this method should always be used as the propeller so obtained is considerably more efficient than the corresponding one from Case 1.

Problem 33

Submarine of the double hull (Lake) type, the propellers being carried under the hull but being given large tip clearances bewteen each other and from the hull. The surface speed to be 16 knots and the submerged speed 13 knots. The effective horse-powers for these speeds being 1030 and 954, total on two shafts.

Hull dimensions:

L.L.W.L. = 221'	Nominal B.C. = .4475
Beam = 23'.5	Twin Screws
H (Surface) = 12'.5	$B \div$ L.L.W.L = .1063
Displ. = 830	Slip B.C. (Surf. and Subm.) = .817 (Line V, Sheet 17).

Surface trials to be run on even trim. Find propeller characteristics, revolutions and power both surface and submerged.

The maximum diameter of propeller possible = 5 ft. 9 in. Maximum surface revolutions not to exceed 375 per minute and minimum submerged revolutions at full submerged power of 1300 S.H.P. (total on both shafts) not to be less than 300 per minute. Propellers three-bladed.

Limiting e.h.p. ÷ E.H.P. (Sheet 22B) for S.B.C. = .817 and $v_1 = 13$, is approximately .9.

$$\text{Subm. Condition } \frac{954}{\text{E.H.P.}} = .9 \quad \therefore \quad \text{E.H.P.} = \frac{954}{.9}. \quad \therefore$$

$$\frac{1030}{\text{E.H.P.}} = \frac{1030 \times .9}{954} = \frac{927}{954} = .972 = \frac{\text{e.h.p.}}{\text{E.H.P.}}, \text{ surface.}$$

SUBMERGED CONDITION

e.h.p. ÷ E.H.P.	.900	.900	.900
v	16	16	16
v_1 (subm.)	13	13	13
$v \div V$	.8 (Min.)	.825	.85
V	16.25	15.76	15.3
e.h.p. (one screw)	477	477	477
E.H.P.	530	530	530
D	5′.75	5′.75	5′.75
(P.A. ÷ D.A.) × E.T.$_p$	2.841	2.929	3.018
P.A. ÷ D.A. (Sheet 24)	.311	.317	.324
P.C.	.675	.672	.669
I.H.P.	785.2	788.7	792.2
Z	−.0477	−.0477	−.0477
I.H.P.$_p$	703.5	706.7	709.8
K	1	1	1
I.H.P.$_d$	703.5	706.7	709.8
(Subm.) S.H.P.$_d$ = I.H.P.$_d$ × .92	647.3	650.1	653.1

POWER—SURFACE

v	16	16	16
e.h.p. ÷ E.H.P.	.972	.972	.972
Z	−.035	−.035	−.035
I.H.P.$_p$	724.4	727.6	730.9
K	1	1	1
I.H.P.$_d$	724.4	727.6	730.9
Surf. S.H.P.$_d$	666.5	669.4	672.4

TO FIND PITCH

T.S.	6880	7000	7130
$1-S.'$	.943	.942	.941
P	4'.585	4'.375	4'.174

ESTIMATE OF REVOLUTION

Log A_V	(Curve X, Sheet 21)	3.63	3.585	3.545
Log A_v	(Sufr. Curve Y, Sheet, 21)	3.44	3.44	3.44
	(Subm. Curve Y, Sheet 21)	3.17	3.17	3.17
s	Surface	.07959	.07472	.06932
	Submerged	.1473	.1351	.1254
R_d	Surface	384.2	400.5	417.3
	Submerged	337	348.1	360.8

These results show that the diameter is too small for straight chart conditions of design and it therefore becomes necessary to resort to either Case 1 or 2 of diameter reduction as in the preceding problem. In solving by these methods use e.h.p. ÷ E.H.P. = .9 for the submerged condition and $v \div V$ for that same condition equal .8.

Problem 33. Double-ended Ferry Boat

Hull Conditions:

Slip B.C. for after propeller = .76

$K = 1.29$

Total e.h.p. = 1122

Per cent e.h.p. delivered by after propeller = $63\frac{2}{3}$

Draft = 13′

Revolutions = 125

$v = 13$ knots

e.h.p. delivered by after propeller = $1122 \times 63\frac{2}{3} = 714$

I.H.P.$_d$ on after propeller = 55 per cent of total power

Maximum $D = 11'$—Propellers four-bladed

Approximate Limits of e.h.p. ÷ E.H.P. for S.B.C. = .76 and $v = 12$, assuming that when in actual service, the vessel may be slowed down to this speed by increased resistance due to overloading and to wind resistance, equal, from Sheet 22B, .36 and .57. Use from .3 to .6.

e.h.p. ÷ E.H.P.	.3	.4	.5	.6
e.h.p.	714	714	714	714
E.H.P.	2380	1785	1428	1190
v	12	12	12	12
$v \div V$	.662	.733	.789	.838
V	18.13	16.37	15.21	14.32
D	11′	11′	11′	11′
Blades	4	4	4	4
(P.A. ÷ D.A.) × E.T.$_p$	2.703	2.245	1.933	1.711
P.A. ÷ D.A. (Sheet 24)	.298	.264	.24	.223
$\frac{4}{3}$ P.A. ÷ D.A.	.397	.352	.32	.297
P.C. for $\frac{4}{3}\ \frac{\text{P.A.}}{\text{D.A.}}$	.622	.651	.67	.682
I.H.P.	3826	2742	2131	1745
Z	.5445	.4144	.3135	.231
I.H.P.$_p$	1092	1056	1036	1025
K (Line C—C, Sheet 19)	1.29	1.29	1.29	1.29
I.H.P.$_d$	1409	1362	1336	1322
Total Power = I.H.P.$_d$ ÷ .55	2562	2476	2430	2404
T.S. for P.A. ÷ D.A.	6610	5820	5250	4830
$1-S$ for $\frac{\text{P.A.}}{\text{D.A.}}$ and S.B.C.	.933	.936	.938	.94
P	10′.29	10′.52	10′.82	11′.05
Log A_V (Cu ve X, Sheet 21)	3.77	3.63	3.54	3.46
v_1	13	13	13	13
Log A_{v_1} (Curve Y, Sheet 21)	3.17	3.17	3.17	3.17
s	.09821	.0917	.09109	.08866
R_d	139	137.8	134	130.9

These revolutions are all too high. In order to obtain the proper number we may proceed in four ways; 1st, by decreasing the values of $v \div V$ until the critical thrusts are reached, values of e.h.p. ÷ E.H.P. constant; 2d, by decreasing the values of $v \div V$ until the critical thrusts are reached, e.t. ÷ E.T. constant; 3d, by Case 1, diameter reduction; 4th, by Case 2, diameter reductions. These methods have already been explained, but in order to obtain a comparison of results from these different methods, we will take one of the above conditions, say the .4 e.h.p. ÷ E.H.P. propeller as a base and depart from it in each of the above ways, as follows:

DOUBLE-ENDED FERRYBOATS

Method.	Constant e.h.p. ÷ E.H.P.			Constant e.t. ÷ E.T.			Case 1, Diam. Red.			Case 2, Diam. Red.		
e.t. ÷ E.T.	.55	.6	.635	.55	.55	.55	.55	.55	.55	.55	.55	.55
e.h.p. ÷ E.H.P.	.4	.4	.4	.4	.37	.33	.4	.4	.4	.4	.4	.4
e.h.p.	714	714	714	714	714	714	714	714	714	714	714	714
D	11′	11′	11′	11′	11′	11′	11′	11′	11′	11′	11′	11′
D_1	11′	11′	11′	11′	11′	11′	11′	12′	13′	11′	12′	13′
$D_1 \div D$	1	1	1	1	1	1	1	1.0909	1.1818	1	1.0909	1.1818
$(D_1 \div D)^2$	1	1	1	1	1	1	1	.....	.....	1	1.19	1.397
$(D_1 \div D)^{3/2}$	1	1	1	1	1	1	1	.....	.....	1	1.14	1.285
$(D_1 \div D)^{1/2}$	1	1	1	1	1	1	1	1.045	1.087	1		
$(D_1 \div D)^{1/4}$	1	1	1	1	1	1	1	.....	.....	1	1.022	1.043
e.h.p.$_1$	714	714	714	714	714	714	714	714	714	714	627	556
E.H.P.	1785	1785	1785	1785	1930	2164	1785	1785	1785	1785	1568	1390
v	12	12	12	12	12	12	12	12	12	12	12	12
$v \div V$	.733	.67	.63	.733	.67	.6	.733	.733	.733	.733	.733	.733
V	16.37	17.91	19.05	16.37	17.91	20.0	16.37	16.37	16.37	16.37	16.37	16.37
Blades	4	4	4	4	4	4	4	4	4	4	4	4
Form	Stand.	Stand.	Stand.	Stand.	Stand.	Stand.	Stand.	Fan	Fan	Stand.	BroadT	BroadT
(P.A. ÷ D.A.) × E.T.$_p$	2.245	2.052	1.929	2.245	2.219	2.228	2.245	1.886	1.607	2.245	1.657	1.252
P.A. ÷ D.A.	.264	.249	.24	.264	.264	.264	.264	.236	.215	.264	.219	.188
$\frac{4}{3}$ P.A. ÷ D.A.	.352	.332	.320	.352	.352	.352	.352	.316	.288	.352	.292	.252
P.C.	.651	.663	.67	.651	.651	.651	.651	.673	.687	.651	.695	.701
I.H.P.$_1$	2742	2692	2664	2742	2965	3324	2742	2652	2598	2742	2289	1983
Z	.4144	.4144	.4144	.4144	.45	.5	.4144	.4144	.4144	.4144	.4144	.4144
I.H.P.$_{p_1}$	1056	1037	1024	1056	1052	1051	1056	1022	1001	1056	882	764
K	1.29	1.29	1.29	1.29	1.29	1.29	1.29	1.29	1.29	1.29	1.29	1.29

$I.H.P._{d_1}$	1362	1338	1321	1362	1357	1356	1362	1318	1291	1362	1137	985
$I.H.P._d$	1362	1338	1321	1362	1357	1356	1362	1438	1526	1362	1354	1376
Total Power	2476	2431	2403	2476	2467	2464	2476	2670	2774	2476	2462	2502
T.S.	5820	5470	5250	5820	5820	5820	5820	5150	4620	5820	4730	3890
$1-S$	.936	.936	.937	.936	.936	.936	.936	.939	.940	.936	.940	.941
P	10'.52	12'.24	13'.55	10'.52	11'.51	12'.86	10'.52	12'.93	15'.64	10'.52	14'.07	18'.51
v_1	13	13	13	13	13	13	13	13	13	13	13	13
Log $A v$	3.63	3.695	3.83	3.63	3.695	3.89	3.63	.363	3.63	3.63	3.36	3.63
Log A_{v_1}	3.17	3.17	3.17	3.17	3.17	3.17	3.17	3.17	3.17	3.17	3.17	3.17
s	.0917	.1049	.1405	.0917	.09767	.1370	.0917	.08740	.08597	.0917	.08597	.08454
R_{d_1}	137.8	120.3	113.2	137.8	126.8	118.7	137.8	111.6	92.39	137.8	102.5	77.75
R_d	137.8	120.3	113.2	137.8	126.8	118.7	137.8	116.6	100.4	137.8	104.7	81.07

PROPELLERS

	$\frac{\text{e.h.p.}}{\text{E.H.P.}}$ Const.	$\frac{\text{e.t.}}{\text{E.T.}}$ Const.	Basic.	Fan.	Basic.	Broad T.						
D	11'	11'	11'.47	11'	11'.55	11'						
R	125	125	122	125		125						
$\frac{4}{3}\frac{\text{P.A.}}{\text{D.A.}}$	.339	.352	.331	.3599	.315	Tip of Basic Cut off to 11' D.						
P	11'.7	11'.72	11'.82	11'.82	12'.36	12'.36						
Total $I.H.P._d$	2446	2467	.....	2550	.....	2461						
P.C.	.462	.458	.....	.443	.....	.459						
Number	1	2	3		4							

An examination of the resultant wheels reveals very small differences between Nos. 1, 2 and 3, except in surface, the power required increasing as the surface increases so that No. 3 promises as the wheel of lowest efficiency. In wheel No. 4, surface has been replaced by pitch and the efficiency is again high. It must be remembered, however, that in the case of twin screw vessels where the propellers are so located that the thrust deduction varies with tip clearance for standard formed blades, but really as clearance of the center of pressure, the thrust deduction to be expected with blades of the forms of No. 3 and No. 4 will be considerably higher than those experienced with standard formed blades, and therefore the efficiency will be less than promised when standard thrust deduction values are used. With single screw vessels it is doubtful whether this augmentation of thrust deduction occurs unless the propeller is roofed over by immersed hull.

Problem 34

Twin screw tunnel boat. Propellers located as shown in Fig. 15. Tip clearance between blades and tunnel roof should not exceed 1 in. Nominal B.C. and Slip B.C. are taken equal to each other $=.8$. K is constant for type and equals 1.195. I.H.P.$_d$ per propeller$=150=300$ total. R_d running free without tow$=225$. Speed running free$=8$ statute miles per hour. Speed when towing$=6$ statute miles per hour. Maximum diameter of propellers$=5$ ft. 6 in. Propellers to be four-bladed.

$$\text{Limits of } \frac{\text{e.h.p.}}{\text{E.H.P.}} \text{ are for}$$

$$\left\{ \begin{array}{l} \text{S.B.C.} = .8 \\ v = \dfrac{8 \times 88}{101.33} = 6.95 \end{array} \right\} = .087,$$

and

$$\text{for} \left\{ \begin{array}{l} \text{S.B.C.} = .8 \\ v = \dfrac{6 \times 88}{101.33} = 5.21 \end{array} \right\} = .057.$$

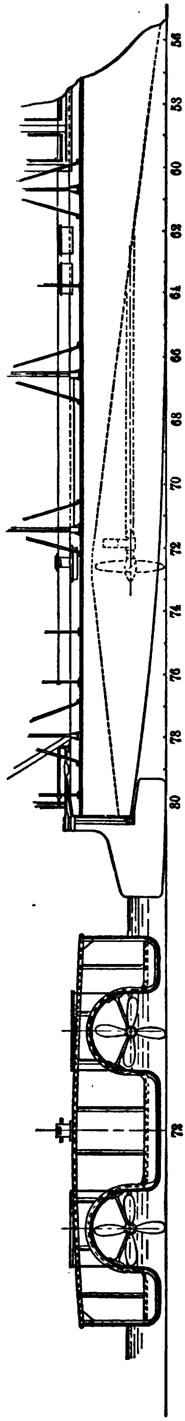


FIG. 15.—Showing Arrangement of Propellers in Tunnel Boats.

As the speeds are very low it will be necessary in order to obtain a practical propeller, to use the free speed and to use a somewhat higher e.h.p. ÷E.H.P. than the lowest limit for the S.B.C. and speed. Let us take $\frac{\text{e.h.p.}}{\text{E.H.P.}}$ =.1 and take what can be realized for towing ability, then

e.h.p. ÷ E.H.P.	.1	.1	.1	.1
$v \div V$ (above *E.T.*)	.36	.4	.44	.48
v (free) (knots)	6.95	6.95	6.95	6.95
V	19.3	17.37	15.79	14.48
I.H P.$_d$	150	150	150	150
K*	1.0973	1.0973	1.0973	1.0973
I.H.P.$_p$	136.7	136.7	136.7	136.7
Z	1.0414	1.0414	1.0414	1.0414
I.H.P.	1504	1504	1504	1504
D	5'.5	5'.5	5'.5	5 .5
I.T.$_D \div (1-S)$	6.416	7.129	7.842	8.555
P.A. ÷ D.A	.398	.422	.446	.468
$\frac{4}{3}$ P.A. ÷ D.A	.531	.563	.595	.624
P.C.	.535	.525	.525	.525
E.H.P.	804.5	789.5	789.5	789.5
e.h.p.	80.45	78.95	78.95	78.95
T.S.	8580	9000	9470	9890
$1-S$	.93	.927	.923	.919
P	4'.235	3'.645	3'.163	2 .788
v (free)	6.95	6.95	6.95	6.95
Log A_V (Curve X, Sheet 21)	3.84	3.71	3.585	3.475
Log A_v (Curve X, Sheet 21	2.655	2.655	2.655	2.655
s	.1069	.08265	.06538	.05339
R_d	182.6	210.5	238.2	266.7

* The proper value to use for K is 1.195.

Laying down P, R_d on $\frac{4}{3}$ (P.A. ÷ D.A.) as abscissas, the propeller to meet the free route condition is found to be as follows:

$D=5'\ 6''$
$P=3'\ 4\frac{1}{2}''$
$\frac{4}{3}\frac{\text{P.A.}}{\text{D.A.}}=.581$
Blades = 4
I H P. = 300 Total on two screws
$v=8$ statute miles
$R_d=225$
e.h.p. = 157.9 by two screws
P C. = .526.

To analyze for the towing condition, proceed as follows:

D	5′.5			
P	3′.375			
$\frac{1}{3}\frac{\text{P.A.}}{\text{D.A.}}$	.581			
P.C.	.525			
$\frac{\text{P.A.}}{\text{D.A.}}$	.436			
T.S.	9250			
$1-S$	.925			
V	16.49			
I T.$_D$	6.95			
I.H.P.	1505			
E.H.P.	790			
v (knots)	5.211	5.211	5.211	5.211
$v \div V$	.3159	.3159	.3159	.3159
$\frac{\text{e.h.p.}}{\text{E.H.P.}}$	.05	.07	.09	.10
$\left(\frac{E.T.}{\text{e.t.}}\right)^z$	1.0	.70	.60	.55
Z	1.355	1.2	1.09	1.0414
K*	1.0973	1.0973	1.0973	1.0973
Log A_V	3.65	3.65	3.65	3.65
Log A_v	2.41	2.41	.241	2.41
$s=S\frac{\text{I.H.P.}_d \times A_V}{\text{I.H.P.} \times A_v}$	.1299	.1299	.1299	.1299
$s=S\frac{\text{K} \times A_V}{10^Z \times A_v} \times \left(\frac{\text{e.t.}}{E.T.}\right)^z$	.06315	.1289	.1938	.2364
R_d	179.8	179.8	179.8	179.8

* K should be 1.195.

Curves of s cross at e.h.p. $\div$ E.H.P. $=.0705$.

∴ e.h.p. delivered per propeller in towing with 150 I.H.P. per engine, at 6 statute miles $=790 \times .0705=55.7$, total $=111.4$.

The problem may be solved by using Cases 1 and 2, "Diameter Reduction" with some possible gain for the towing condition. The broadening of the blade tips will, however, undoubtedly injure the performance in free route due to the increase in thrust deduction.

Motor Boats

In the design of propellers for motor boats, curves of effective horse-power and curves of full-throttle engine power, when the engine carries a varying brake load, plotted on revolutions,

should be furnished as if when the estimated power-revolution curve of performance of the propeller is laid down, the curve of full-throttle-revolution curve of the engine should fall below it at any point, the engine would not be able to carry the revolutions above this point and disappointment in speed would result.

Also in boats of this class where the speed is so great that the vessel planes, the point where this planing begins will be shown on the effective horse-power curve by a decided hump in the curve. The standard block coefficient should be found in the ordinary manner by Sheet 17, up to the vertex of the hump. From this point on the slip block coefficient rapidly decreases, until at a speed equal to about $\frac{3}{2}$ times the speed at the hump the slip block coefficient will equal about 50 per cent of the standard slip block coefficient.

CHAPTER X

DESIGN OF PROPELLERS BY COMPARISON

Sometimes, in designing the propellers for a vessel, it is desired to obtain propellers which will give an equal propulsive efficiency with those fitted to an earlier vessel of similar form but of different size, and whose performance has been regarded as excellent.

In the method of comparison here proposed, the formulas take the following forms:

L_1 = Length of original vessel;

L_2 = Length of new vessel;

(Both on L.W.L.)

$$r=\frac{L_2}{L_1};$$

$$D_2=D_1r^{3/2};$$

$$P_2=P_1r^2;$$

$$R_2=R_1\div r^{3/2};$$

$$V_2=V_1r^{1/2};$$

$$\text{Apparent slip}_1=\frac{P_1\times R_1-101.33\ V_1}{P_1\times R_1};$$

Apparent slip$_2$ =

$$\frac{P_1r^2\times\frac{R_1}{r^{3/2}}-101.33\ V_1r^{1/2}}{P_1r^2\times\frac{R_1}{r^{3/2}}}=\frac{P_1\times R_1-101.33\ V_1}{P_1\times R_1}.$$

$\therefore$ Apparent slip$_1$ = Apparent slip$_2$;

$$\text{Tip-speed}_1 = R_1 \times \pi D_1;$$

$$\text{Tip-speed}_2 = R_2 \times D\pi_2 = \frac{R_1}{r^{3/2}} \times \pi D_1 r^{3/2} = R_1 \times \pi D_1;$$

$\therefore$ Tip-speed$_1$ = Tip-speed$_2$;

Again,

$$\text{I.H.P.}_2 = \text{I.H.P.}_1 r^{7/2};$$

$$\text{Disc area}_2 = \frac{\pi}{4} D_1^2 r^3;$$

$$\text{Disc area}_1 = \frac{\pi}{4} D_1^2;$$

$$P_2 \times R_2 = P_1 r^2 \times \frac{R_1}{r^{3/2}} = P_1 r^{1/2} \times R_1;$$

I.T. per square inch disc area$_2$

$$= \frac{\text{I.H.P.}_1 \times r^{7/2} \times 33{,}000}{P_1 r^{1/2} \times R_1 \times \frac{\pi}{4} D_1^2 r^3} = \frac{\text{I.H.P.}_1 \times 132{,}000}{P_1 \times R_1 \times \pi D_1^2};$$

I.T. per square inch disc area$_1$

$$= \frac{\text{I.H.P.}_1 \times 132{,}000}{P_1 \times R_1 \times \pi D_1^2}.$$

$\therefore$ I.T.$_1$ = I.T.$_2$, and for the model screw the tip-speed, apparent slip, and thrust per square inch of disc area are identical with those of the original screw.

The equations for Diameter, Pitch and Revolutions may also be put in the following forms:

To obtain such propellers, it is possible to work directly with the Chart formulas, always remembering that, according to the Charts, for equal propulsive efficiencies the projected-area ratios, and products of thrusts (effective, propulsive, and indicated), by tip-speeds must remain equal. Bearing this in mind,

and obtaining the ratios between the equations for diameter, pitch, and revolutions of the existing propellers, and those of the propellers that are being designed, the following equations result:

$$D_2=D_1\sqrt{\frac{\text{E.H.P.}_2\times V_1}{\text{E.H.P.}_1\times V_2}}=D_1\sqrt{\frac{\text{I.H.P.}_2\times V_1}{\text{I.H.P.}_1\times V_2}}=D_1\sqrt{\frac{\text{S.H.P.}_2\times V_1}{\text{S.H.P.}_1\times V_2}};$$

$$P_2=P_1\sqrt{\frac{\text{E.H.P.}_2\times V_2}{\text{E.H.P.}_1\times V_1}}=P_1\sqrt{\frac{\text{I.H.P.}_2\times V_2}{\text{I.H.P.}_1\times V_1}}=P_1\sqrt{\frac{\text{S.H.P.}_2\times V_2}{\text{S.H.P.}_1\times V_1}};$$

$$R_2=R_1\sqrt{\frac{\text{E.H.P.}_1\times V_2}{\text{E.H.P.}_2\times V_1}}=R_1\sqrt{\frac{\text{I.H.P.}_1\times V_2}{\text{I.H.P.}_2\times V_1}}=R_1\sqrt{\frac{\text{S.H.P.}_1\times V_2}{\text{S.H.P.}_2\times V_1}}.$$

Where

D_1 = Diameter of existing propeller;
D_2 = Diameter of propeller for new ship;
P_1 = Pitch of existing propeller;
P_2 = Pitch of propeller for new ship;
R_1 = Revolutions of existing propeller;
R_2 = Revolutions of propeller for new ship;
V_1 = Speed of existing vessel;
V_2 = Speed of new ship;
E.H.P.$_1$ = Effective horse-power for V_1 of existing vessel;
I.H.P.$_1$ = Indicated horse-power for V_1 of existing vessel·
S.H.P.$_1$ = Shaft horse-power for V_1 of existing vessel;
E.H.P.$_2$ = Effective horse-power for V_2 of new vessel;
I.H.P.$_2$ = Indicated horse-power for V_2 of new vessel;
S.H.P.$_2$ = Shaft horse-power for V_2 of new vessel.

The speeds used in the above should be the corresponding speeds by Froude's Law of Comparison, where

$$V_2=V_1\left(\frac{\text{Displacement}_2}{\text{Displacement}_1}\right)^{1/6},$$

and

$$\text{Power}_2=\text{H.P.}_2=\text{H.P.}_1\left(\frac{\text{Displacement}_2}{\text{Displacement}_1}\right)^{7/6},$$

the following forms will obtain:

$$D_2 = D_1\left(\frac{\text{Displacement}_2}{\text{Displacement}_1}\right)^{1/2} = D_1\left(\frac{L_2}{L_1}\right)^{3/2} = D_1\left(\frac{V_2}{V_1}\right)^3$$
$$= D_1\sqrt{\frac{\text{H.P.}_2 \times V_1}{\text{H.P.}_1 \times V_2}};$$

$$P_2 = P_1\left(\frac{\text{Displacement}_2}{\text{Displacement}_1}\right)^{2/3} = P_1\left(\frac{L_2}{L_1}\right)^2 = P_1\left(\frac{V_2}{V_1}\right)^4$$
$$= P_1\sqrt{\frac{\text{H.P.}_2 \times V_2}{\text{H.P.}_1 \times V_1}};$$

$$R_2 = R_1\left(\frac{\text{Displacement}_1}{\text{Displacement}_2}\right)^{1/2} = R_1\left(\frac{L_1}{L_2}\right)^{3/2} = R_1\left(\frac{V_1}{V_2}\right)^3$$
$$= R_1\sqrt{\frac{\text{H.P.}_1 \times V_2}{\text{H.P.}_2 \times V_1}},$$

where L_1 and L_2 are the load water line lengths of the old and the new vessel respectively.

According to these formulas it appears that the "Law of Mechanical Similitude" does not apply to screw propellers, as the diameters are seen to vary approximately as the cubes of the speeds, while the pitches vary, with the same degree of approximation, as the fourth power.

CHAPTER XI

EFFECT ON PERFORMANCE OF THE PROPELLER CAUSED BY VARYING ANY OF ITS ELEMENTS

Effect of Change of Blade Form on Performance

Should the forms of projected areas here advocated not be adhered to, the following results may be confidently looked for:

1. **Broadening the Blades at the Tips.** Revolutions will be decreased, apparent slip will be decreased, and thrusts will be increased and efficiency slightly decreased.

2. **Narrowing the Blades at the Tips.** Revolutions will be increased, apparent slip increased, and thrusts decreased.

In the matter of relative weights for equal blade strengths the narrow-tipped blade has the advantage.

It should be distinctly understood that no claim is made that the forms advocated in this work are necessarily those giving the maximum efficiency. It is believed that equal efficiencies can be obtained with all shapes, if for each shape the proper diameter, pitch, and surface have been provided for the absorption of the delivered power under the conditions in which the screw is operating. Each series of forms must, however, have its own particular factors of design if results in conformity with the computed performance are to be expected.

Some Points Governing Propulsive Efficiency

1. **Effect of Excess Pitch.** Shown by Fig. 16.—Gain in propulsive coefficient at low powers. Loss in propulsive coefficient at high powers. Both sets of propellers having blades exactly alike, but projected area ratio decreasing as pitch increases.

2. **Effect of Variation of Blade Surface.** Least surface: Greatest efficiency at low powers; rapid loss of efficiency as

power increases; least efficiency and earlier cavitation at high powers. Blades all of same form, which was the standard form. Maximum surface: Greatest efficiency and smoothest running

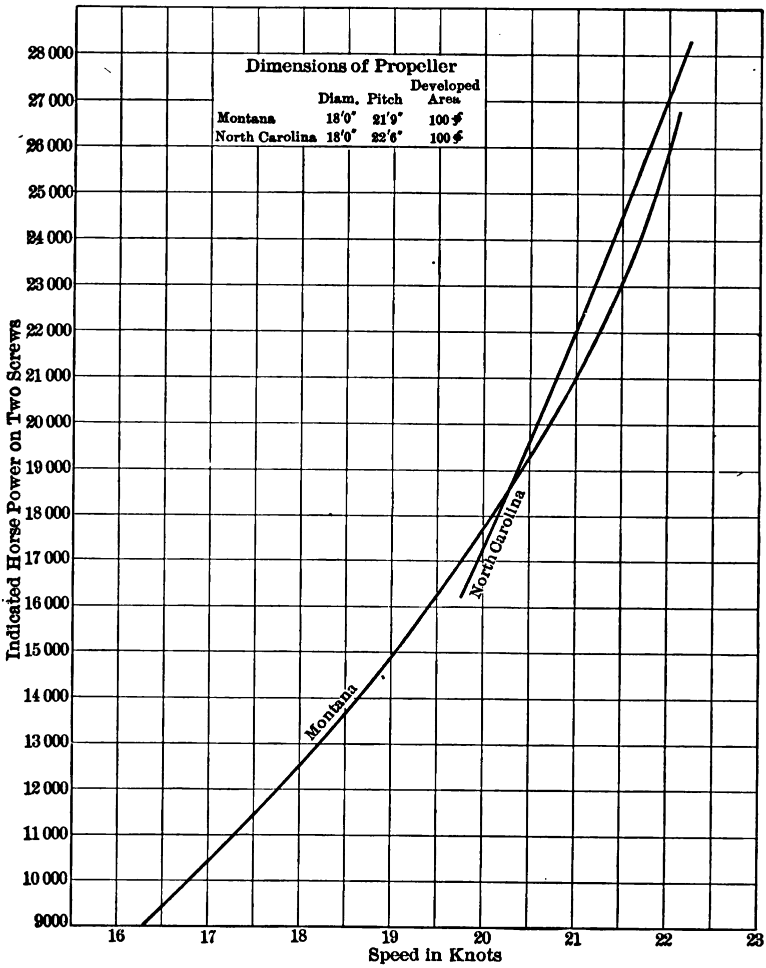


FIG. 16.—Influence of Projected Area Ratio on Efficiency.

at high powers; lowest efficiency at low powers. Lowest tip-speeds for equal indicated thrusts per square inch of disc area with the other screws. See Fig. 17.

3. **Effect of Variation of Power Distribution on Four-shaft Installation.** One H.P. ahead turbine on each outboard (wing) shaft. One L.P. ahead, one backing and one M.P. cruising

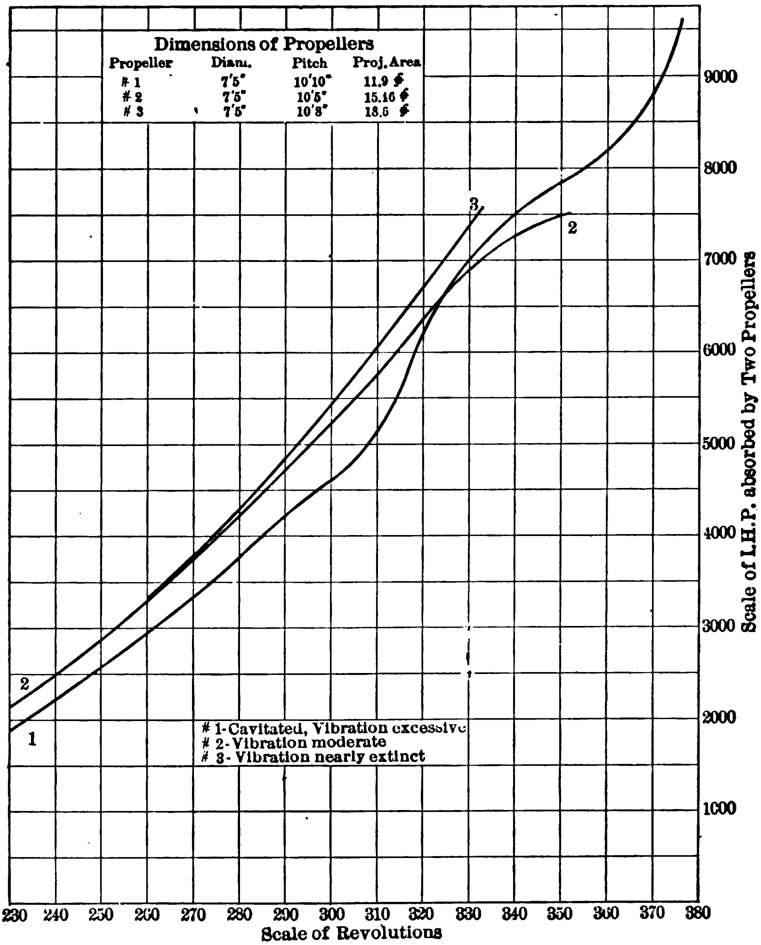


FIG. 17.—Power and Revolutions as Affected by Projected Area Ratio.

turbine on one inboard shaft. One L.P. ahead, one backing and one H.P. cruising turbine on other inboard shaft. See Fig. 18. These variations will vary with the distribution of power on the shafts.

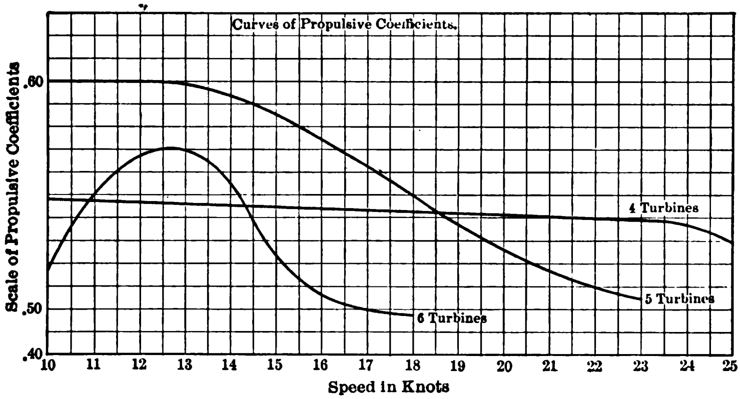


FIG. 18.—Effect of Varying Distribution of Power on Four-shaft Arrangement.

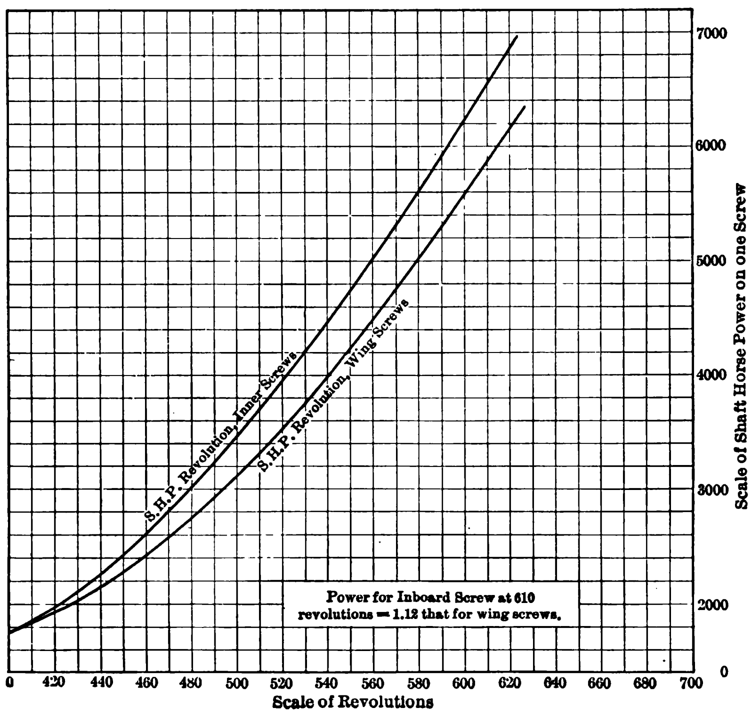


FIG. 19.—Effect of Position of Propellers on Power and Revolutions, Four-shaft Arrangement.

4. **Four-shaft Arrangement.** All propellers of the same dimensions. Effect of position of propellers in relation to the hull on the power and revolutions. See Fig. 19.

5. **Cases A and B.** Three-shaft arrangements.

Case A. Dead wood cut away. Center propeller working in locality well clear of hull.

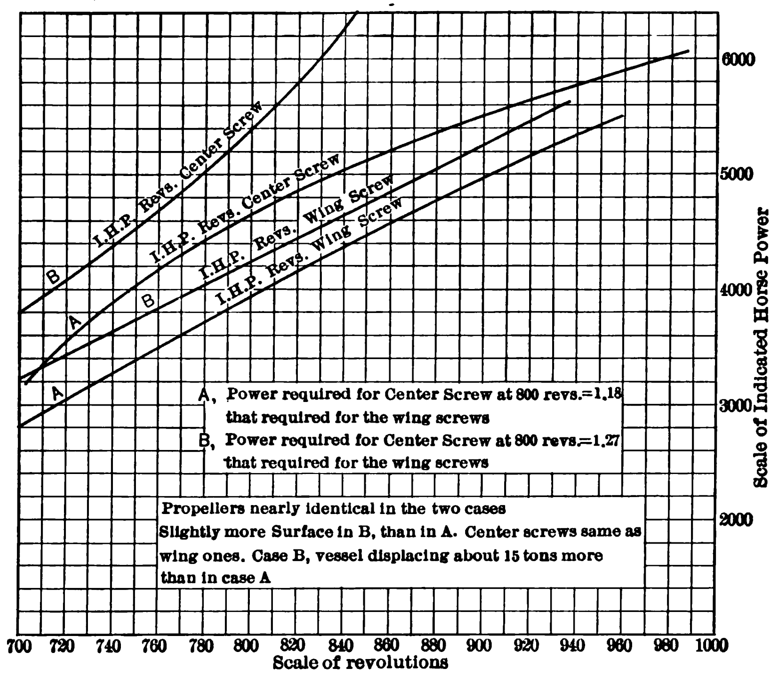


FIG. 20.—Three-shaft Arrangement. Influence of Character of Afterbody on Power and Revolutions.

Case B. Dead wood carried well aft. Center propeller working immediately in wake of stern post.

Center propellers of same dimensions as wing ones and propeller of Case A almost identical in dimensions with Case B. See Fig. 20.

6. **Effect on Propulsive Efficiency of Location of Propeller When Operating in the Wake of a Full Afterbody.** An interesting problem of the above conditions has recently arisen

in the case of a self-propelled barge constructed for the Navy Department. The block coefficient of the hull was .9, the afterbody being very full.

The propeller, as first fitted, was located as shown by the dotted lines in Fig. 21, the hull lines shown being those of the actual vessel.

The contract speed of the barge was six knots, but, although

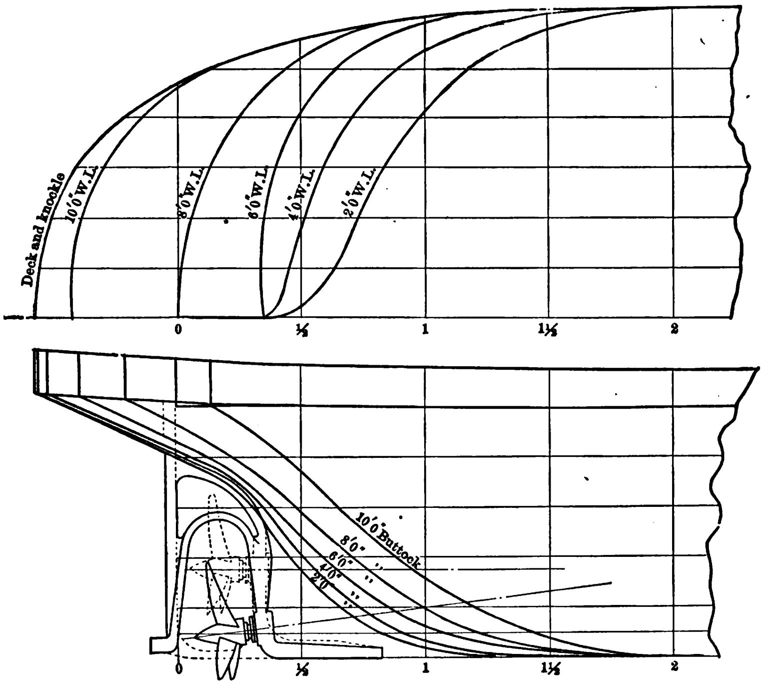


FIG. 21.—Positions of Maximum and Minimum Efficiency Positions of Propeller in the Case of Very Full Afterbodied Shallow-draft Vessel.

a series of seven propellers was tried, the maximum speed obtained was only approximately $5\frac{1}{4}$ knots.

After thoroughly considering the conditions, it was determined that the best chance of success, at the least expense, was offered by relocating the propeller so that a better flow of water to it from forward would occur. This idea was adopted, and the propeller was located as shown by the full lines of Fig. 21, the shaft being given a very heavy inclination.

After this change had been made, the vessel was again tried and a speed of approximately $6\frac{1}{4}$ knots was realized with about the same power and revolutions that had given $5\frac{1}{4}$ knots under the original conditions.

The results of the various trials and the data of the propellers used are given in Table IX, the trial marked No. 8 being the final successful one. The major part of improvement in performance in this case was caused by the change in location of the propeller, the second position permitting a much freer and more direct flow of water to the propeller, a much reduced thrust deduction factor resulting therefrom.

TABLE IX

U.S.N. OIL BARGES NOS. 2 AND 3, COURSE ON CHESAPEAKE BAY

Prop. Nos	1	2	3	4	5
Date	9-27-11	10-5-11	10-7-11	10-13-11	10-19-11
Diameter of wheels	5′ 6″	6′ 2″	5′ 9″	6′ 3″	6′ 2″
Pitch	4′ 3″	3′ 3″	3′ 6″	3′ 8″	3′ 6″
Number of blades	4	3	3	3	4
Dev. area square feet	12.92	10.0	6.58	8.58	13.33
Proj. area square feet	11.72	.942	.60	8.0	12.13
Average steam	111.25	110	107.5	115.2	122.6
R.P.M.	204.3	212.9	210.5	208.2	200.75
Slip per cent	49	30.4	36.4	30.8	23.8
I.H.P.	133.78	149.92	130.95	144.74	152.24
Speed knots	4.415	5.117	4.695	5.10	5.136

Prop. Nos	5*	6	7	8
Date	10-24-11	11-6-11	12-20-11	5-8-12
Diameter of wheels	6′ 2″	6′ 3″	6′ 3″	6′ 9″
Pitch	3′ 6″	3′ 0″	3′ 4″	3′ 6″
Number of blades	4	3	4	3
Dev. area square eet	13.33	15.21	20.28	14
Proj. area square feet	12.13	14.64	19.52	13.2
Average steam	123.7	125	125	129.53
R.P.M.	207.602	228.65	205.95	206.27
Slip per cent	26.7	26.15	22.5	12.56
I.H.P.	160.0	172.67	175.47	177.64
Speed knots	5.258	5.045	5.24	6.235

Propeller number marked * is *Official Trial.*

No. 8. Line of shaft so modified as to bring lower blades of propeller well below keel of vessel.

CHAPTER XII

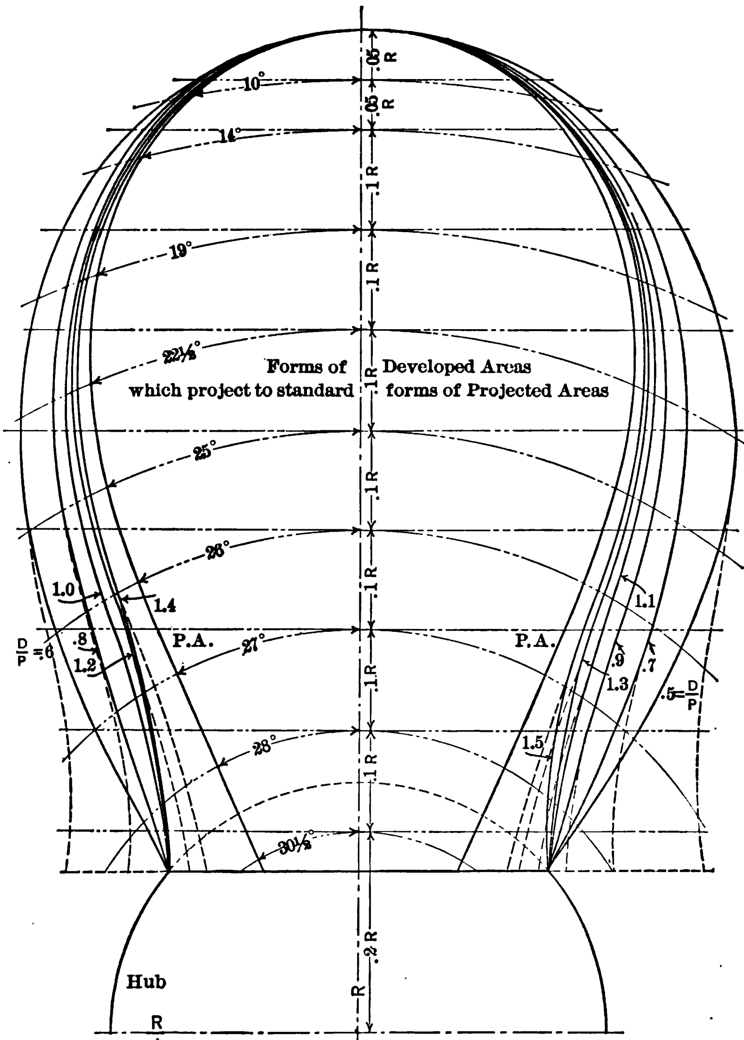
STANDARD FORMS OF PROJECTED AREAS OF BLADES FOR USE WITH THE CHARTS OF DESIGN

Forms of Blades and Blade Sections

Returning to the Barnaby presentation of Froude's results, the constants obtained by Mr. Barnaby are only correct so long as the ratio of developed area to disc area and the elliptical form of this developed area used by him are adhered to. There is no way of allowing for the effect of increase or decrease in this developed area except the rough one of estimating that the total thrust that can be delivered by the propeller will vary directly as the developed area.

By investigation of what occurs when the standard elliptical blade used by Barnaby in his experiments is broadened or narrowed, it is readily seen that this method of correcting for variations in developed area ratio is incorrect, for as the blade widens, for any one pitch ratio, the length of its resistance arm increases above that of the standard width blade, and as it narrows the length of this arm decreases. In the first case, the resistance of the blade to turning is increased not only by the increased surface friction of the larger blade, but also by the increase in the length of the radius to the center of pressure of the blade. Should the blade be narrowed below standard, the opposite effect will be produced.

If elliptical blades of varying pitch ratios, but having the same area of projection on the disc, be laid down, it will be noted that the form of projection, not only on the disc but also on the plane of the axis of the propeller, changes in passing from the lower to the higher pitch ratios. This same change in form of projected area also occurs if blades of the same pitch ratio but of



Projected area = .32 disc area
Dotted lines are developments of projected areas as shown
Full " " actual developments of blades
For other projected area ratios increase or decrease projected area shown proportionally on circular arcs

FIG. 22.—Developments Corresponding to a Standard Projected Area Ratio of Constant Value but with Varying Values of Pitch ÷ Diameter.

different values of developed area ratio are laid down, thus in both cases showing that not only the resistance of the blade, due to change of surface, and in the first case change of pitch, has been affected, but it has been still further modified by the change in the distribution of this surface and the modification in the leverage arm of the blade resistance.

A distinguished educator who formerly was an advocate for the use of constant developed area form in design work, has put before the public a work on propellers in which he advocates the use of the projected area in place of developed area. He has adopted, instead of the constant elliptical form of development, a constant elliptical form of projected area. Should the projected area ratio be .3 its form is that of an ellipse, and should this ratio be .6 the form is again an ellipse. The same changes in distribution of surface occur as before, and no benefit has been obtained except that of having an easy form to lay down and also one that can be mathematically represented.

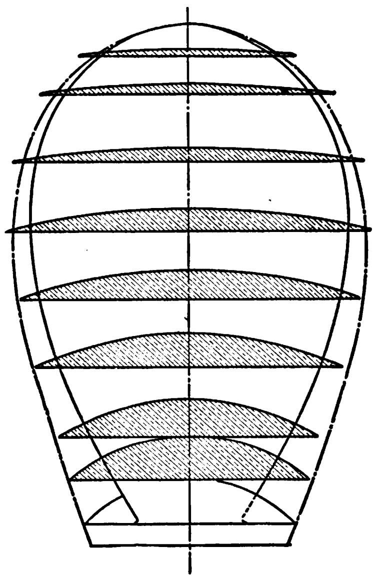


FIG. 23.—Projected Area Standard Form and Blade Sections.

In order to maintain as constant a distribution of blade area as possible, and thus guard against changes in resistance due to changes in distribution, there has, in this work, been adopted as a basic form of projected area the form of projection of the blades of two 3-bladed propellers having approximately a projected area ratio of .32. These two propellers had most excellent records, which could hardly be bettered.

Using a standard hub diameter equal to .2 of the diameter of the propeller, this standard basic form of projection was drawn.

Then, with the center of the hub as a center, and different radii, circular arcs were struck crossing the axis of the projection. In obtaining the projected forms for areas differing from the basic area, the widths of the projections measured on these circular arcs were made proportional to the circular arc widths of the basic projection; that is, a .6 projected area ratio would have circular arc measurements $\frac{60}{32}$ times as great as those for the basic .32 projection.

The forms of projected area so obtained, when compared with the forms of blades of many propellers, are found to agree very closely, from the lowest to the highest values of P.A. ÷ D.A., with those forms which have the best records credited to them.

By using these forms, for any pitch ratio the resistance arm of the blade always remains the same, no matter what the developed or projected area ratio, and the only change in resistance to turning is that caused by the additional surface. The forms of projection, both on the disc and on the fore-and-aft plane, remain constant for all pitch ratios.

Naval Constructor D. W. Taylor, in writing on the effect of blade form as deduced from model tank experiments, stated "A good practical rule would seem to be to make the blades broader at the tips for low-pitch ratios and narrow them for high ones." The blades here advocated follow this rule automatically, as the broadest part of the developed blade, measured on the elliptical development of the circular projected arc, moves slowly in towards the hub as the pitch ratio increases, thus gradually narrowing the tips of the blade for the higher pitch ratios.

The derived projected forms are shown on Sheet 25 (Atlas), where is also shown a diagram by which the developed area ratio can be obtained for standard propellers of any pitch ratio and any given projected-area ratio, and vice versa. On this same sheet is given a table of multipliers for obtaining the lengths of chords for half circular arc widths for different projected-area-ratio blades for any desired diameter of propeller, by means of which the projected-area forms can be laid down without the use of the diagram of forms.

The necessity of adhering to standard projected-area forms in

order to obtain graphical or other methods of design will be easily understood by examining the following figures, 24, 25, and 26, showing forms of projected and developed areas for the Standard, the Barnaby, and the Taylor forms of blades.

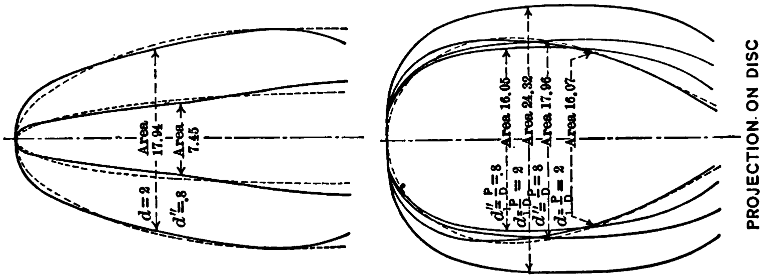


FIG. 26.—Taylor Form.

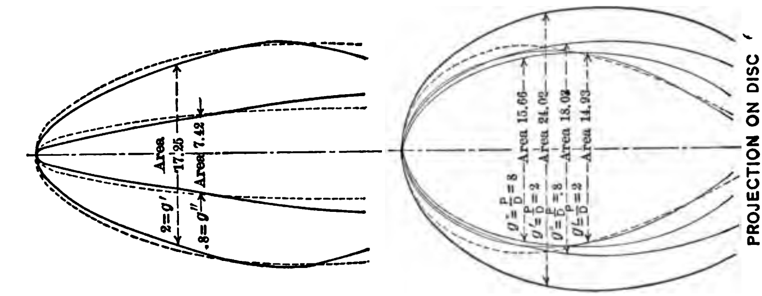


FIG. 25.—Elliptical Form.

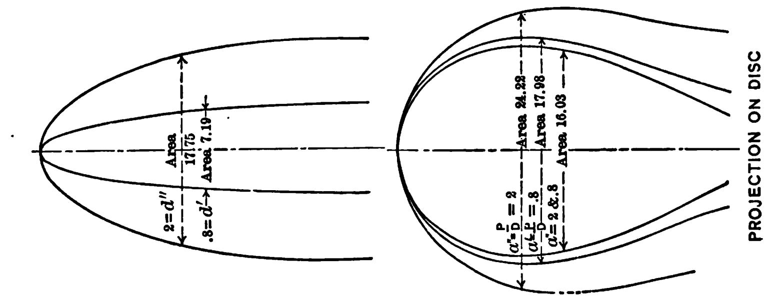


FIG. 24.—Standard Form.

Fig. 24 shows the standard form of projected area marked a, and the developed areas for pitch ratios of .8 and of 2, marked a' and a'', respectively, together with the corresponding projections on the fore-and-aft plane.

Figs. 25 and 26 show equal developed areas with Fig. 24, and the corresponding projected areas on the disc and on the fore-and-aft plane, for Barnaby's and for Taylor's blades, respectively.

The dotted forms shown on Figs. 25 and 26 are the projected-area forms of Fig. 24.

Attention is called to the great variation in distribution of blade surface, as the pitch ratio changes, in blades of the forms given in Figs. 25 and 26, and the rational and gradual change of form that occurs in Fig. 24.

The standard form need not be adhered to rigidly, but may be modified between the greatest width and the hub in order to decrease the resistance of the sections of the blade in this region by allowing the blade to be made wider, thinner, and sharper-edged.

Variations from the Standard Form

With the exception of the last-mentioned case there are but two cases where departures from the standard form are justified, after such form has been adopted and the method of design been based upon it. These cases occur when limitations of draught or conditions of design make it impossible to fit a propeller of as large a diameter as is indicated by the calculations to be necessary to obtain the desired revolutions with the maximum possible efficiency.

Such cases are shown by projected areas shown in Figs 4 and 5, and as A and B in Fig. 27, having a diameter of screw of $2R$. With case A, the allowable radius is R', so, while retaining the pitch of the greater diameter propeller, it becomes necessary to broaden the tip of the blade which takes the form shown by A', the area of the projection A' being equal to the projected area A. This area A' may be as shown, or may be greatly modified in appearance, as in A'', provided the circular arc measurements of width at equal radial distances remain equal.

Where the difference between the calculated and the allowed diameters is large, the resultant blade would have an abnormal

form, as in B'. This form is often met with in motor-boat propellers, disguised as B''. Patents have been allowed on this form, and great claims are made for it on the grounds of high efficiency, when in reality its greater efficiency over a blade of ordinary form is caused by its approximation in amount and

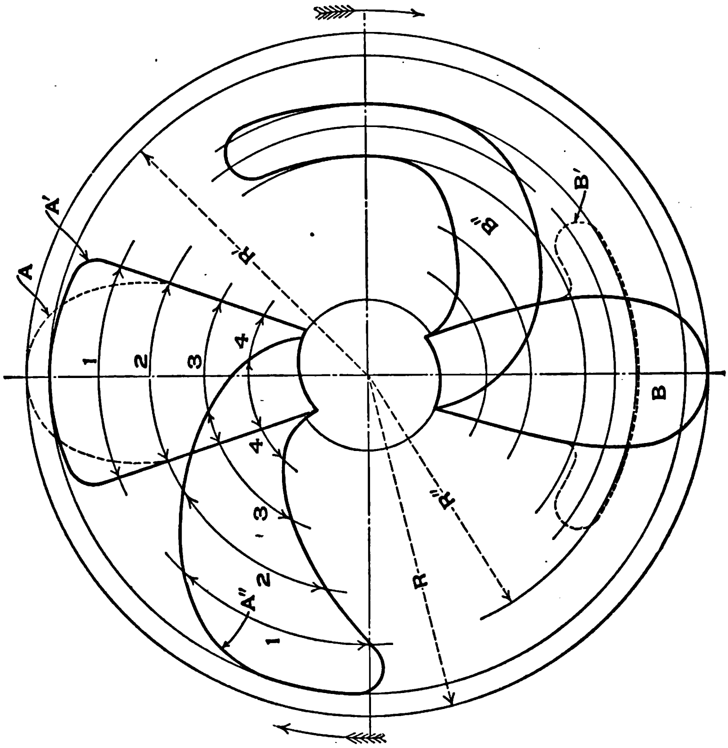


FIG. 27.—Variations in Projected Area Forms.

distribution of area to those of a propeller having the proper dimensions for the work which it is called on to do. In place of this tip broadening, however, the problem may be solved for the diameter that can be carried, and the standard form of projected area be adhered to. Blades having the greatest width thrown well out beyond its location in the standard form

are about 3 per cent less efficient for the same projected-area ratio.

Blades thrown to the side, as in A'', are used to reduce vibration in cases where a rapid-running screw operates close to the strut or stern post. As a general rule, though, they are undesirable, as the form is weak and the blade must be made extra heavy at the root in order to provide the necessary rigidity to insure against change of pitch, due to springing of the blade when subjected to heavy thrusts.

Rake of Blades. It is a very common practice to rake the blades aft to a more or less degree, and this practice was generally followed in the United States Navy until a few years ago. There was a generally accepted idea that centrifugal action of the screw was decreased and that efficiency was increased by so doing.

An examination of the performances of actual screws in service, and of model screws in the tank, shows that there is no solid ground for either belief. In the cases of the actual screws, no difference in the propulsive efficiencies of screws with and screws without rake can be noticed, and the models gave practically identical results. As to centrifugal action, numerous tank experiments have shown the propeller race to be almost cylindrical, and that so far from there being centrifugal action, there appears to be a slight convergence abaft the propeller as shown by Fig. 1 A.

An actual advantage gained by raking the blades aft is that the blade tips of wing screws are given greater clearance from hulls of usual form than if the blades were radial; also, for the same blade clearance, the strut arms may be made shorter. Another advantage which the rake may have is in giving greater clearance between the leading edges of the blades and the after side of the stern post and struts, this additional clearance allowing the water a chance to enter the disc at a better angle.

Radial blades, in addition to being as efficient as those with rake, are more easily machined, have less total developed area for equal projected area, and therefore less surface friction, are stiffer and lighter; also, the stresses in the blades due to cen-

trifugal action are less. With propellers of high speed of revolutions, this latter point is very important. and for such screws the blades should never rake.

Form of Blade Sections for Standard Blades

In the propellers designed according to the Dyson method the form of section existing in the propellers from whose data the design data curves were developed has been adhered to. In these blades, the working face of the blade forms the nominal pitch surface, the blades in all cases being made with constant pitch. The thickness of the blade is built up on the back of nominal pitch face. The form of the back is an arc of a circle and the edges are made as thin and sharp as possible without sacrificing durability to an extravagant degree. See Fig. 23.

The principal forms of blade section that are met with in practice are as follows:

With a small value of $\frac{T}{W}$ from .12 to .20 at the hub, the form *A* appears, from trials, to be all that is required. Where the value of $\frac{T}{W}$ is higher than .20 and the fillet of the blade is also heavier it may be advisable to slightly fine the entrance of the blade by throwing back the leading edge a small amount as shown in *B*, but this should not be done to any great extent, as it tends to slow the blade down by increasing the actual working pitch above the nominal more than is done by *A*.

Section *C* with the following edge of the blade thrown back, the leading edge being either similar to *A* or thrown back as shown, is considered to be a decided mistake, since, as the water travels along the driving face of the blade from the entering to the leaving edge, there may exist a tendency for it to break contact with the blade face. It was to guard against this tendency of the water to leave the blade face that axially expanding pitches of blades were used. If the following edge of the blade is thrown back as in *C*, the face of the blade is deliberately drawn away from the water and a cavity at this edge will result, with conse-

quent eddying effect and resultant vibration and loss in efficiency.

In Section D, the leading half of the back of the blade has a pitch such that its slip equals the real slip of the screw. This form is theoretically correct, provided the velocity of the water meeting the propeller is that of the vessel modified according to the wake that is equal to $v-w$, but in practice, unless the blades

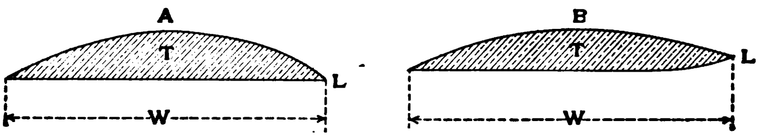


FIG. 28.

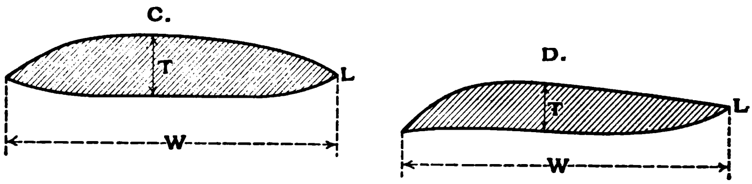


FIG. 29.

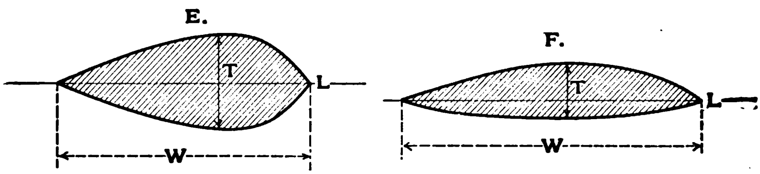


FIG. 30.—Variations in Blade Sections.

are very wide, it gives too thick a blade and too blunt an entrance, with a consequent heavy loss in efficiency.

Blades of section E, with the metal of the blade divided evenly on each side of the nominal pitch surface or plane, appear to offer less resistance to turning than any of the other sections, due probably to the fact that the real pitch of the blade is probably the same as the nominal pitch, as is shown by the fact that if blades of this section, designed for zero pitch, be revolved, they will exert zero thrust, while blades of the preceding sections,

designed for zero pitch, will, when revolved, record a decided thrust, due to the influence of the backs of the blades; the back evidently giving the blade a working pitch greater than the nominal pitch causes it to exert a thrust. While it requires less power to turn blades of section *E*, the resultant thrust per revolution is much lower and the apparent slip is much higher than with blades of the same nominal pitch but of different section.

With blades having sections similar to *F*, the same results are obtained as with Section *E*, but in a less degree. The design curves and factors being based on the performances of blades of manganese bronze, it is desirable, when a weaker material is used, to make the thickness from the pitch plane to the back the same that it would be if the stronger material were used, and to add the additional thickness to the face, thus producing a section similar to *F*.

CHAPTER XIII

THICKNESS OF THE BLADE AT ROOT. CENTRIFUGAL FORCE. FRICTIONAL RESISTANCE OF PROPELLER BLADES

THICKNESS OF THE BLADE

THE fiber stress to be used in determining the thickness of the blade at the root depends upon the material of which the blade is to be made and the degree of approximation of the point of design to full overload conditions. The material usually used for propellers in the Naval Service is manganese bronze, and the specified ultimate tensile strength of the material is 60,000 lb. Where the possible overload does not exceed 10 per cent, a working stress of 10,000 lb. per square inch can be used with safety with reciprocating engines. With turbines or reduction gear this may be increased to 13,000 lb. This is the condition existing for Sheet 20 of the Design Sheets. For propellers designed for about .3 load factor, $\frac{\text{e.h.p.}}{\text{E.H.P.}}=.3$, where the power used in the calculations may be very much lower than the maximum power possible, this working stress should be reduced to about 6000 lb. For high speed, high-powered motor boat propellers, the thickness with high-grade material may be made $\frac{3}{4}$ in. for each foot radius of propeller. Plate A.

The formula used for the determination of blade thickness has been derived from Naval Constructor D. W. Taylor's work on "Resistance of Ships and Screw Propulsion," and is an adaptation of the formula proposed by him. The nomenclature and formulas are as follows:

T = Thickness of blade at and tangent to hub, additional thickness due to fillets being neglected. T should not exceed $.2W$.

W = Width of blade tangent to hub.

$A = (33{,}000 \times \text{I.H.P.}_d) \div (2\pi \times \text{Revolutions} \times \text{Number of blades}) = 5252\ \text{I.H.P.}_d \div (R \times N)$ = Maximum indicated torque per blade, in foot pounds.

$B = .31 \times$ Diameter of screw, in feet = Mean arm.

$C = A \div B$ = Resultant athwartship force on one blade, in foot pounds.

$D = 12 \times B -$ Radius of hub, in inches = Arm of athwartship force measured to root of blade.

$E = C \times D$ = Athwartship moment at root of blade, in inch pounds.

$F = (33{,}000 \times \text{I.H.P.}_d) \div (\text{Pitch, in feet} \times \text{Revolutions} \times \text{Number of blades})$ = Indicated thrust per blade, in pounds.

$G = .345 \times$ Diameter of propeller, in inches = Mean arm of thrust, in inches.

$H = G -$ Radius of hub, in inches = Arm of thrust measured to root of blade, in inches.

$J = F \times H$ = Fore-and-aft moment at root of blade, in inch pounds.

K = Circumference of hub, in feet $\div$ Pitch, in feet = Tangent of angle between face of blade and center line of hub or fore-and-aft line tangent to surface of hub.

L = Sine of arc whose tangent is K.

M = Cosine of arc whose tangent is K.

$N = L \times J$ = Component of fore-and-aft moment normal to face of blade at root.

$O = M \times E$ = Same for athwartship moment.

$P = N \div O$ = Total moment at root of blade in inch pounds.

f = Fiber stress = as per values of e.h.p. $\div$ E.H.P. given on Plate A.

$$T = \sqrt{\frac{P \times 13.125}{W \times f}}.$$

Fixing the maximum thickness at $T = .2W$, T should never exceed $T = \sqrt[3]{\frac{P \times 2.625}{f}}$ for the strong bronzes.

For cast iron and semi-steel, f = from 2500 to 4000, for values of e.h.p. $\div$ E.H.P. not in excess of .4.

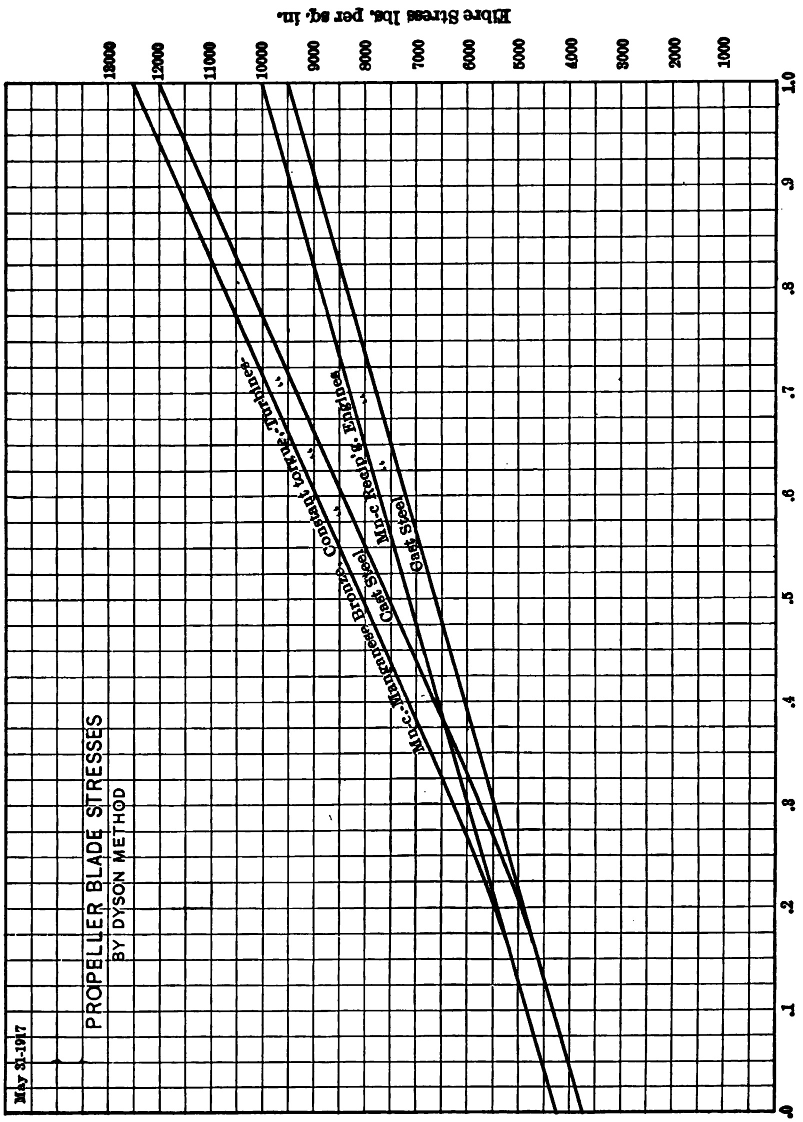


PLATE A.—Curve of Fiber Stresses on Propeller Blades.

Centrifugal Force: Increase of Stress

Concerning the effect of this on blade stress, Seaton states, " Centrifugal force produces in the screw blade at all times some stress, and at high revolutions the stress becomes serious, so much so, in fact, that destruction of blades is due sometimes to this source with screws driven by turbines.

" Within moderate velocities the forces set up by inertia really tend to balance those by hydraulic pressure on the blade. That is to say, that whereas the hydraulic action tends to bend the blade in a direction opposite to that of revolution, the inertia of the blade tends to make it bend the other way as well as to ' throw off.'

" The forces acting on a screw blade due to its velocity can be calculated from the usual formula where W is the weight of a blade in pounds, r is the distance of its center of gravity from the axis of rotation, g is gravity and taken at 32, v the velocity in feet per second:

Then

$$C=\frac{W}{g}\times v^2,$$

and the tension on the bolts $=\frac{W}{g}\times\frac{v^2}{r}$,

$\frac{v^2}{r}$ being of course the accelerating force, and called usually the centrifugal force.

" When a propeller is in motion on normal conditions, running at R revolutions per minute,

$$v=\sqrt{\text{pitch}^2+(2\pi r)^2}\times R\div 60=\frac{R}{60}\sqrt{P^2+(2\pi r)^2}.$$

" As an example take the case of a screw propeller 12 ft. diameter, 15 ft. pitch, 200 revolutions per minute; center of gravity of blade is 3.2 ft. from the center; it weighs 1600 lb.

Determine the bending moment on the root distance 1.8 ft. from the *c.g.* and the tension on the screw bolts screwing it to the boss.

$v=\sqrt{\overline{15}^2+(2\pi\times3.2)^2}\times200\div60=\frac{10}{3}\sqrt{\overline{15}^2+(2\pi\times3.2)^2}=84$ feet. per second.

$C=\frac{1600}{32}\times\overline{84}^2=352,800$ lb.

"Tension on bolts $=352,800\div3.2=110,125$ lb.

"If seven bolts, tension on each $=15,732$ lb.

"Bending moment due to $C=352,800\times1.8=635,040$ ft.-lb. This is, however, in a plane through the face at the *c.g.* and therefore is resisted by the section at the root longitudinally.

"Taking circular motion and no advance of the screw,

$$v=2\pi\times3.2\times200\div60=67 \text{ ft.}$$

Then

$$C=\frac{1600}{32}\times\overline{67}^2=224,500 \text{ lb.}$$

"The bending moment on a plane at right angles to axis $=224,500\times1.8=404,100$ ft. lb.

"Taking an extreme case of an Atlantic steamer driven by turbines so that each screw receives 18,000 I.H.P. at 180 revolutions, the diameter being 16 ft. 6 in., the pitch 18 ft., the weight of each blade 11,200 lb., its *c.g.* being 4.5 ft. from the axis and 2.0 ft. from the root.

"Here velocity $=\frac{180}{60}\sqrt{18^2+(\pi9)^2}=111$ ft. per second.

$$C=\frac{11,200}{32}\times\frac{\overline{111}^2}{2240}=1925 \text{ tons.}$$

"Taking circular velocity only,

$$C=\frac{11,200}{32}\times\frac{\overline{85}^2}{2240}=1129 \text{ tons.}$$

"Tension on bolts $=\frac{1129}{4.5}=251$ tons.

"If thirteen bolts to each blade, the load on each $=19.3$ tons in addition to that due to the pressure on the blade."

Frictional Resistance of Propeller Blades

The following method of estimating the frictional resistance of propeller blades is given by Mr. A. E. Seaton in his work on "Screw Propellers."

"Frictional resistance of a screw blade may be found by the following simple methods: Fig. 31 shows the outline of the developed surface of half a blade whose figure is symmetrical about *CB*. The propeller is moving at a uniform rate of revolution so that *BC* represents the velocity through the water at the tips to a convenient scale.

"That is, the velocity per revolution at *B* and at any intermediate point is

$$v_1=\sqrt{\overline{\text{pitch}}^2+(\pi d)^2}=\sqrt{P^2+(\pi d)^2}$$

d being the diameter at any point taken.

"If *BC*, etc., *GK*, represents on a convenient scale the velocities at *B*, etc., *G*. A curve drawn through *C*, etc., *K* will permit of the velocity being ascertained at any intermediate points by taking the intercept between *BG* and *CK* at these points. The resistance per square foot may be calculated at three or four points by the rule $y=1.25\left(\frac{V}{10}\right)^2$ lb. and a curve *GD* set up in the same way so that intercepts will give the resistance at any intermediate points."

Now, taking narrow strips of the blades at three or four stations and multiplying by the resistance at these stations and doubling the result to allow for the blade backs, a curve *HE* is obtained so that intercepts again give the resistance at various stations, and the area is the measure of the total resistance of one blade.

Proceed, then, to multiply the resistance of the strips as obtained above by the space moved through by them in a minute, and the work absorbed in turning the blade is measured by making a curve *HF* by means of a few of the ordinates so found as before.

Intercepts between *HF* and *GB* give the work absorbed in moving those strips through the water, and the area *GBFH* represents the total power in foot pounds absorbed in turning that blade through the water.

Dividing it by 33,000, the horse-power required to overcome *h* is obtained.

Fig. 31 represents the equivalent resistance of two of the four blades of H.M.S. *Amazon*, and Fig. 32 is that of one of the

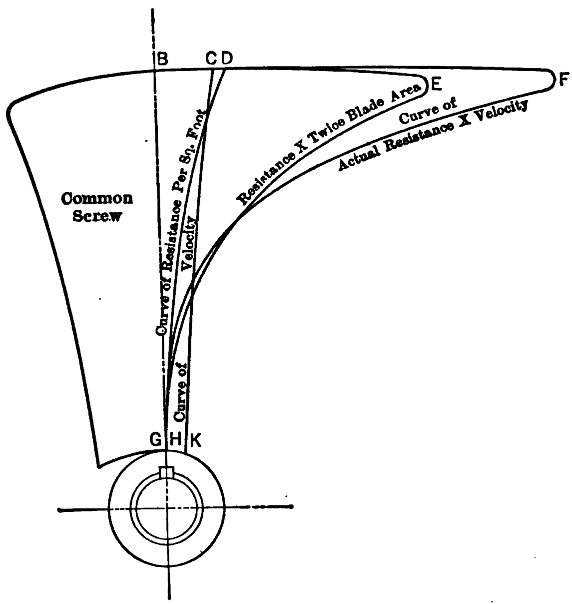


FIG. 31.—Estimate of Blade Resistance.

two of the Griffiths screw which replaced it and gave so much better results.

The ill effect of the broad tip is seen at a glance, as are also the losses arising from excessive diameter, for, by taking 6 in. off each tip, the resistance is in both cases very much reduced, especially so in the case of the four-bladed screw. Froude found the efficiency of the *Greyhound's* machinery to be exceedingly low, and attributed it chiefly to engine resistance, whereas it was largely due to the absurdly large diameter of the screw,

it being 12.33 ft. diameter with 52 sq. ft. of surface; whereas the *Rattler*, of similar size and power, had a screw 10 ft. diameter with only 22.8 sq. ft. of surface, which elaborate experiments years before had shown to be sufficient. Moreover the *Rattler* had a speed coefficient (Admiralty) of 224 against that of 142 of the *Greyhound*, which should have opened the eyes of the authorities in 1865.

With the high speed of revolution necessary for the efficient working of turbine motors, as also for the speed of revolution possible with modern reciprocating engines, especially the enclosed variety with automatically forced lubrication, propellers of small diameter are absolutely necessary for safe running, while to prevent cavitation the blade area must be relatively large. Hence it is found that the modern propeller is gradually getting nearer and nearer in width of blade to the common screw of sixty years ago, and differs from it now chiefly in its having nicely rounded corners instead of the rigidly square ones of that time.

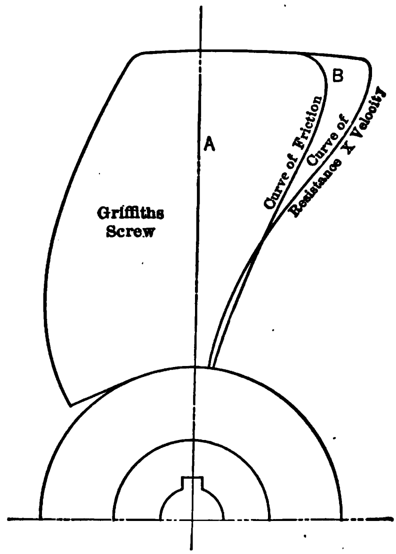


FIG. 32.—Estimate of Blade Resistance.

Fig. 33 shows one blade of H.M.S. *Rattler* of 1845; the dotted line is that of a blade of a modern turbine motor steamer. Now, although the difference in blade is small to look at, the action when at work is very different. The corners of the old screws caused violent vibration at high speeds; but when they were cut away there was a very marked improvement.

Frictional resistance of a screw propeller may be calculated with a close approximation to the truth by taking the velocity at the tip and the total area of acting surface, using multipliers in

both cases deduced from the close calculation of it with screws of different types.

Let V be the velocity of the blade tips in knots per hour.

Let R be the revolutions per minute.

Let D be the diameter in feet.

Let P be the pitch of screw in feet.

Let A be the area of acting developed surface in square feet.

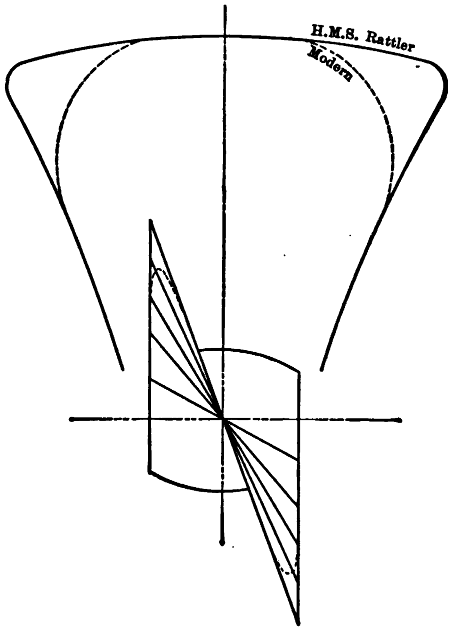


FIG. 33.—Antique and Modern Propeller Blades.

The resistance of a square foot is assumed to be $1\frac{1}{4}$ lb. at 10 knots.

$$V=\frac{60\times R}{6080}\sqrt{P^2+(\pi D)^2}=\frac{R}{101.33}\sqrt{P^2+(\pi D)^2}.$$

$$\text{Resistance per square foot}=1.25\left(\frac{V}{10}\right)^2 \text{ lb.}$$

$$\text{Resistance of screw}=2A\times 1.25\left(\frac{V}{10}\right)^2\times f \text{ lb.}$$

For a common screw.................. $f = .6034$
For a fantail-shape screw............. $f = 0.581$
For a parallel blade.................. $f = 0.550$
For an oval......................... $f = 0.520$
For a leaf shape..................... $f = 0.450$
For a Griffiths....................... $f = 0.350$

The horse-power absorbed in overcoming the frictional resistance may be found now by multiplying the resistance by the space in feet moved through in a minute and dividing by 33,000.

The mean space moved through by the blade surface from tip to boss of an ordinary propeller $= 0.7 \times$ distance moved through by the tip.

Hence mean space $= \dfrac{V \times 0.7 \times 6080}{60} = 70.9V.$

$$\text{Then I.H.P. expended} = 2A \times 1.25\left(\frac{V}{10}\right)^2 \times f \times 70.98V \div 33{,}000$$

$$= \frac{A \times V^3 \times f}{18{,}612}.$$

Edge resistance $= N \times 5$ per cent $\times$ I.H.P. expended where $N =$ number of blades.

Total resistance of screw = I.H.P. expended + Edge resistance.

Example. A screw 12 ft. diameter, 15 ft. pitch, has 42 sq. ft. of surface and moves at 130 revolutions per minute (three leaf blades).

$$* \text{Here } V = \frac{130}{101.33}\sqrt{225 + 1440} = \frac{40.8 \times 130}{101.3} = 52.4.$$

$$\text{Frictional resistance screw H.P.} = \frac{42 \times \overline{52.4}^3 \times 0.515}{18{,}612} = 166.8.$$

Edge resistance here will be 3×5 per cent or 15 per cent of $166.8 = 25$ H.P.

Then total resistance of screw $= 166.8 + 25 = 191.8$ H.P.

* π^2 is here taken $= 10$.

CHAPTER XIV

CHANGE OF PITCH. THE HUB. LOCATION OF BLADE ON BLADE PAD. DIMENSIONS OF THE HUB

VERY often, upon the trial of a vessel, results indicate that improvement is possible if the propeller blades be set to a higher or lower pitch than that of the designed driving face. In order to provide for such a change, the bolt holes in the blade pads are made oval, so that ordinarily with large blades the blades can be twisted to mean pitches of about 1 ft. more and 1 ft. less than the designed pitch, the new pitch becoming a variable one. If the blade is set for a higher pitch than the designed, the new pitch becomes a radially expanding one, increasing from the hub towards the tips, while if the new pitch is lower than the designed, the pitch will decrease radially from the hub to the tips.

The change caused by alterations in pitch may be obtained from the following table (Table X), (see Peabody's "Naval Architecture"), by multiplying the original pitch ÷ diameter by the factors given in Table X for the small angle through which the blade is twisted.

THE HUB

In designing screw propellers it was, up to the advent of the turbine, the custom almost invariably to design propellers of large diameter with the blades detachable from the hub in order that injured blades might be replaced at little expense, and also that improvement in propulsive efficiency might be sought for by providing for slight modifications of pitch in securing the blades to the hub. Only in the smaller wheels were the blades cast solid with the hubs. With the pitch ratios ordinarily in use with the comparatively high pitch, slow-turning reciprocating-engine propellers, where the hub diameters varied from 20 per cent to 28 per cent of the diameter of the screw in built-up wheels, the pitch angles at the hub ranged

from about 50° to 58°, while with solid propellers with hubs varying in diameter from $14\frac{3}{4}$ per cent to $18\frac{1}{4}$ per cent of the diameter of the propeller, the pitch angle varied from 67° to 76° at the hub.

TABLE X

Angle the Blade is Twisted.		1°		2°		3°		4°		5°		6°	
Pitch Ratio.	Diameter Ratio.	Increase.	Decrease.	Increase.	Decrease.	Increase.	Decrease.	Increase.	Decrease.	Increase.	Decrease.	Increase.	Decrease.
0.8	1.25	1.07	0.93	1.15	0.85	1.22	0.78	1.29	0.71	1.37	0.64	1.45	0.57
0.9	1.11	1.07	0.94	1.14	0.87	1.20	0.81	1.27	0.75	1.34	0.68	1.41	0.61
1.0	1.00	1.06	0.94	1.12	0.88	1.19	0.82	1.25	0.76	1.31	0.70	1.38	0.65
1.1	0.90	1.05	0.94	1.11	0.89	1.17	0.84	1.23	0.78	1.29	0.73	1.35	0.68
1.2	0.83	1.05	0.94	1.10	0.89	1.16	0.84	1.21	0.79	1.27	0.74	1.32	0.70
1.3	0.77	1.05	0.95	1.10	0.90	1.15	0.86	1.21	0.81	1.26	0.76	1.31	0.72
1.4	0.71	1.05	0.95	1.09	0.91	1.14	0.86	1.20	0.82	1.24	0.77	1.30	0.73
1.5	0.66	1.05	0.96	1.09	0.91	1.14	0.87	1.19	0.83	1.24	0.78	1.29	0.74
1.6	0.62	1.04	0.96	1.09	0.91	1.13	0.87	1.18	0.83	1.22	0.79	1.28	0.75
1.7	0.59	1.04	0.96	1.08	0.92	1.13	0.88	1.17	0.84	1.22	0.80	1.27	0.76
1.8	0.55	1.04	0.96	1.08	0.92	1.13	0.88	1.17	0.84	1.21	0.81	1.26	0.77
1.9	0.52	1.04	0.96	1.08	0.92	1.12	0.88	1.16	0.85	1.20	0.81	1.25	0.78
2.0	0.50	1.04	0.96	1.08	0.92	1.12	0.88	1.16	0.85	1.20	0.81	1.25	0.78
2.1	0.47	1.04	0.96	1.08	0.92	1.12	0.88	1.16	0.85	1.20	0.82	1.24	0.79
2.2	0.45	1.04	0.96	1.07	0.93	1.11	0.89	1.15	0.85	1.20	0.82	1.24	0.79
2.3	0.43	1.03	0.96	1.07	0.93	1.11	0.89	1.15	0.86	1.19	0.83	1.24	0.79
2.4	0.42	1.03	0.96	1.07	0.93	1.11	0.89	1.15	0.86	1.19	0.83	1.24	0.80
2.5	0.40	1.03	0.96	1.07	0.93	1.11	0.90	1.15	0.86	1.19	0.83	1.23	0.80

Desirable as it is to reduce the diameter of the hub to that of the strut boss in order to avoid eddying between the boss and strut, it is not always possible to do this with the hubs of built-up propellers. The seating of the propeller-blade pad in the hub must be circular to permit of pitch adjustments, the hub must be spherical to maintain its symmetry of outline when variations of pitch are made, and the seating must be of sufficient diameter to accommodate a proper number of holding bolts of sufficient strength; finally, the blade pad must be of sufficient width to

accommodate a blade having such a ratio of thickness to width as will prevent excessive blade resistance.

The effect of the above requirements, when met for turbine-driven propellers of large diameter and low-pitch ratio, was to throw the effective blade areas too far out from the axis of the hub, thus leading to serious increase in blade friction at the tip-speeds employed, and also to bring the 45° pitch angle of the helical surface well within the surface of the hub. In the few cases coming to notice, in which detachable blades were used for these high-speed turbine screws, the results obtained were poor, but there were other conditions existing in these cases which may have been responsible for the poor propulsive efficiency realized.

Location of Blade on Blade Pad

In order to provide sufficient space for the blade bolts to pass through the pad without cutting into the true working face or

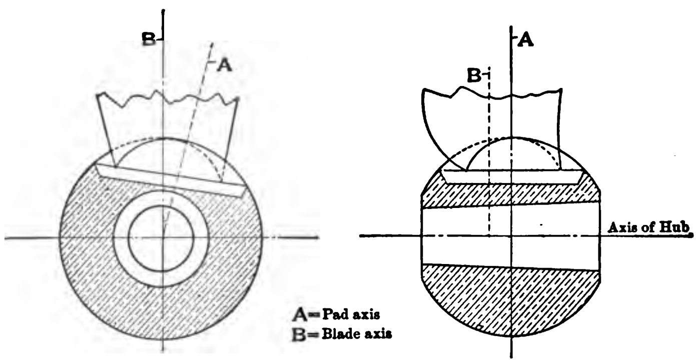


Fig. 34.—Correct Method of Changing Blade Position to Permit Bolting to Hub.

the working section of the blade, disregarding the fillet, it is very often necessary to shift the blade on the pad so that the blade axis does not coincide with the axis of the pad. To do this, the blade axis may be shifted forward or aft of the pad axis along the axis of the hub, or it may be swung around the hub to an angular position with the axis of the pad, or the two shifts may be combined (Fig. 34).

Whatever is done in this way, care must be taken that the axis of the generatrix shall always remain coincident with the axis of the hub. Many cases are encountered where, in order to give sufficient space for the blade bolts, the blade has been moved to one side or the other of the pad axis, the axis of the blade remaining parallel to the axis of the pad, but the axis of the generatrix becoming transferred several inches to one side of the hub axis to which it remains parallel.

Dimensions of the Hub

The following rules for hub dimensions are abstracted from Bauer's "Marine Machinery" and are on the lines of accepted good practice:

Diameter of shaft $= d$.

Propeller Hub. 1. In smaller propellers the hub and blades are cast in one. Length of hub, $L = 2.3$ to $2.6d$. Maximum diameter of hub $d_n = 2.0$ to $2.3d$. Slope of cone of the propeller shaft 1 in 10 to 1 in 16.

As a rule the center part of the hub around the shaft is cut away, firstly, to effect a saving of weight, secondly, to facilitate the fitting of the propeller on to the conical end of the shaft. In order that the turning moment of the shaft may be transmitted to hub at its thickest part, the latter must, especially at the thicker end of the cone, fit accurately on to the shaft. The hub is prevented from turning on the shaft by one or two strong keys.

Key: Breadth of the $b = \frac{d}{6} + \frac{5}{8}$ inch. Thickness of key 0.5 to $0.6b$, d being the diameter of the propeller shaft. If there are two keys, only $0.8b$ instead of b is required. The keys must fit both hub and coned shaft accurately at the sides, but a little clearance may be allowed in the hub at the top. The hub is first fitted on to the shaft without the keys, then removed, and the keys fitted to the shaft in countersunk keyways. The hub is replaced, and it should be possible to push it as far up the cone as before the keys were fitted.

The keys almost always extend the whole length of the hub; but sometimes, if the propeller is small, they occupy only the front half. The propeller nut has a fine thread, and may be made with either indentations or projections.

Diameter of the nut $d_1 = 1.4$ to $1.5d_a$. Thickness of nut $h_1 = 0.75$ to $0.85d_a$, d_a being the diameter over the thread. The smaller values may be used for larger nuts. These values hold for nuts where the shaft has a diameter measured outside the thread of over $5\frac{3}{4}$ in., otherwise d_1 is taken from the table of dimensions of bolts and nuts, and equals the width across the flats of a hexagonal nut. To prevent the nut slacking back, it is usually made with a left-handed thread for a right-handed screw propeller, and a right-handed thread for a left-handed screw propeller, but this rule is often departed from. Some method of locking the nut is also usually provided. To screw on the nut easily, the shaft is continued for a short distance beyond the nut, and given a diameter slightly less than that at the bottom of the thread.

2. *Hubs with Blades Bolted on.* In merchant vessels with propellers over 10 to 13 ft. diameter, and in warships with propellers over 6 ft. 6 in. to 8 ft. 6 in. diameter, the blades may be bolted on to the hub.

In the best practice, the flanges of the blades are very carefully fitted to the surfaces on the hub, to prevent the water getting underneath them, and sometimes a rubber ring is inserted, and screwed up against the hub.

Thickness of flange of blade $f_1 = 0.18$ to $0.22d$ for bronze or cast steel.

Diameter of flange of blade $D_1 = 1.9$ to $2.3d$.

Corresponding to this diameter of flange, the external diameter of the hub is:

d_n 2.6 to $3d$ for large screws;

d_n 3.0 to $3.5d$ for small screws.

Length of hub with blades bolted on, $L = 2.1$ to $2.6d$ (higher values are for smaller hubs).

Thickness of hub round the cone:

$c_1 = 0.19$ to $0.22d$ for bronze;

$c_1 = 0.18$ to $0.21d$ for cast steel;

$c_1 = 0.22$ to $0.24d$ for cast iron.

Thickness of metal at front and back ends of hub:

$w_1 = 0.22d$ for bronze;

$w_1 = 0.20d$ for cast steel:

$w_1 = 0.24d$ for cast iron.

In all these formulæ d is the diameter of the propeller shaft.

CHAPTER XV

STOPPING, BACKING AND TURNING SHIPS

The data given in this chapter were principally obtained and the text prepared by Commander S. M. Robinson, U. S. N., in connection with the development of electric drive for ships. The performance of an induction motor is vitally affected by the performance of the ship so that in addition to the normal "steaming-ahead" condition there are three others that must be considered. These are (1) stopping (that is motors running free with no power on them), (2) backing (with ship going ahead), (3) turning. In the past, little attention seems to have been paid to these points, so it was necessary to do considerable experimenting in order to determine what actually happens in each of these cases.

Stopping

In the case of a ship fitted with reciprocating engines, when the signal "stop" is received the engines are held stopped; if it is necessary, the links are thrown over and enough steam admitted on the backing side to hold the engines. In this case the screws act as a powerful brake, and stop the ship rapidly. In the case of turbine ships, some engineers shut steam off of the ahead turbines and let the propellers keep revolving ahead while others admit steam to the backing turbines to hold the screws stopped; however, it is not believed that the latter practice is much used. An electrically propelled ship is similar to the turbine ship when the latter uses no steam in the backing turbine. The retardation of the speed of a reciprocating-engined ship will therefore be considerably more rapid than that of either a turbine ship or an electrically propelled ship when the engines are stopped.

For the purpose of determining what this retardation is, experiments were conducted on the U.S.S. *Jupiter* by running over a measured course with power off and propellers running freely. Observations were taken on shore and also on board ship, and from these were plotted speed and r.p.m. retardation curves.

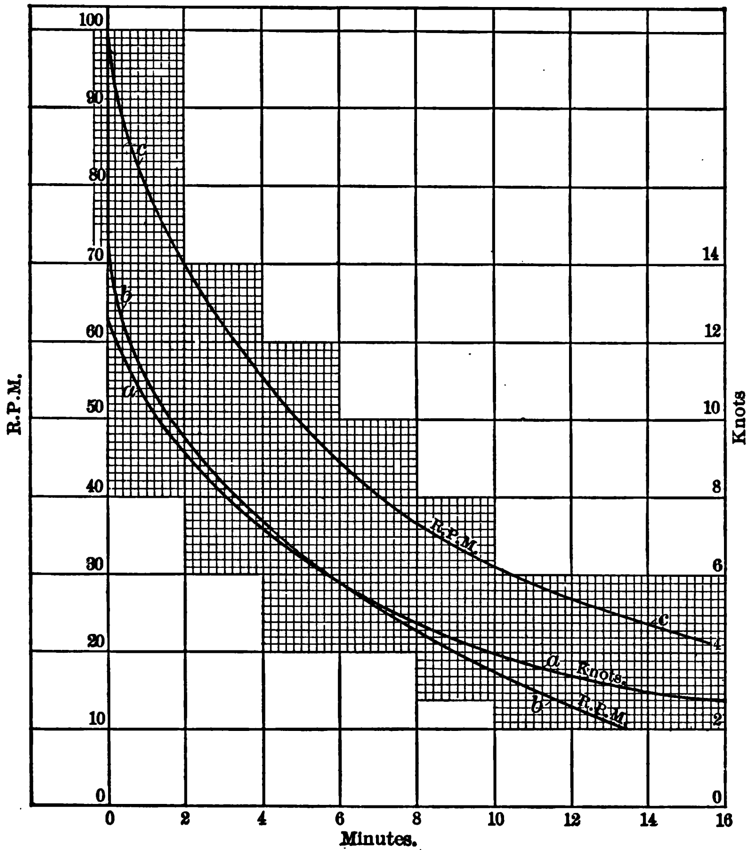


FIG. 35.

These are shown in Fig. 35. Curve *a* represents the actual speed of the ship at any time interval, curve *b* represents actual revolutions per minute of the screws at any time interval, and curve *c* represents the revolutions per minute necessary to drive the ship at the speeds represented by curve *a*. From curve *b* it will be

seen that the apparent slip of the screws, when dragging, is about 28 per cent and from curve c that it is about 9 per cent when going ahead. Apparent slip when going ahead is taken to be,

$$s=\frac{(P\times R)-(v\times 101.33)}{P\times R}.$$

Slip when dragging is taken to be

$$s=\frac{(v\times 101.33)-(P\times R)}{v\times 101.33},$$

where s = per cent slip;
v = speed of ship, in knots;
P = pitch of screw, in feet;
R = revolutions per minute of screw.

This retardation curve shows that the zero torque point on the propeller occurs at about 68 per cent of the r.p.m. necessary to drive the ship; that is to say, the propellers, when dragging, will turn at about 68 per cent of the r.p.m. necessary to drive ahead. Later on it will be seen that this agrees fairly well with results obtained in the model tank, where the zero torque point on the propeller was found to be between 70 per cent and 78 per cent of the r.p.m. necessary to drive the ship.

This retardation curve is necessary for properly working out a retardation curve when applying backing power. As it is not always feasible to actually determine this curve by actual experiment, a method has been worked out for calculating it, and it is believed that it will be accurate enough for all practical purposes.

By this method the retardation curve can be obtained whether the ship is running with engines stopped or backing. The following is the method used:

Let H.P. = horse-power exerted at any instant to stop the ship;
W = work done (per second) by this horse-power;
F = force in pounds acting on the ship to stop it;

M = mass of the ship;
a = retardation in knots per hour per minute;
a' = retardation in feet per second per second;
g = force of gravity = 32.16;
v = speed of the ship, in knots;
Δ = displacement, in tons.

Then

$$W=\frac{\text{H.P.}\times 33{,}000}{60},$$

also

$$W=F\times\frac{v\times 6080}{60\times 60},$$

or

$$\frac{\text{H.P.}\times 33{,}000}{60}=\frac{F\times v\times 6080}{60\times 60};$$

$$\therefore\ F=\frac{\text{H.P.}\times 33{,}000\times 60}{v\times 6080}.$$

But

$$F=Ma'=\frac{\Delta\times 2240}{32.16}\times a',$$

then

$$\frac{\text{H.P.}\times 33{,}000\times 60}{v\times 6080}=\frac{\Delta\times 2240\times a'}{32.16},$$

$$\therefore\ a'=\frac{\text{H.P.}\times 33{,}000\times 60\times 32.16}{v\times\Delta\times 2240\times 6080},$$

$$a=\frac{\text{H.P.}\times 33{,}000\times 60^4\times 32.16}{v\times\Delta\times 2240\times 6080^2}.$$

For the *Jupiter* Δ was 16,670 tons at the time of the experiment, so for that ship

$$a=\frac{\text{H.P.}\times .009964}{v}.$$

To calculate the H.P. acting to stop the ship at any speed v, there must be added together the effective horse-power necessary to drive the ship at the given speed and the horse-power due to the braking effect of the screw if the ship is running with power shut off or the power delivered by the engines if she is backing. If the ship is running without power it is believed that the following method of estimating the braking effect of the screw will be accurate enough. Consider the action of the screw (while revolving freely) to be similar to that of the struts. This seems a reasonable assumption, as the screws will at all times have water back of them if they are revolving. As an example of this method take the *Jupiter:*

Strut area (one)	8.96 sq. ft.
Propeller hub area (one)	7.92 sq. ft.
Total area covered by strut and propeller hub	16.88 sq. ft.
Projected area of propeller (one)	60.56 sq. ft.

From Sheet 18, Atlas, the strut resistance is found to be 9.3 per cent of the resistance of the bare hull.

$\therefore$ Propeller resistance $=\frac{60.56}{16.88}\times 9.3=33.7$ per cent of bare hull resistance.

From Sheet 18, the total appendage resistance is found to be 11.3 per cent of bare hull resistance.

Total added resistance will be

11.3 per cent $+$ 33.7 per cent $=$ 45 per cent.

$\therefore$ Total H.P. $=1.45\times$ effective horse power (bare hull).

In Table XI the values of a have been calculated by substituting these values of H.P. in the equation previously derived. The derived values of a are shown in Fig. 36. In this figure it will be seen that the curve between any two speeds differing by only one knot is nearly a straight line, so that the average retardation, while the ship is dropping one knot, can be taken as

the average of the retardation at the two speeds. Using this method, the time for the ship to drop to any speed has been calculated in Table XI. This gives a retardation curve which has

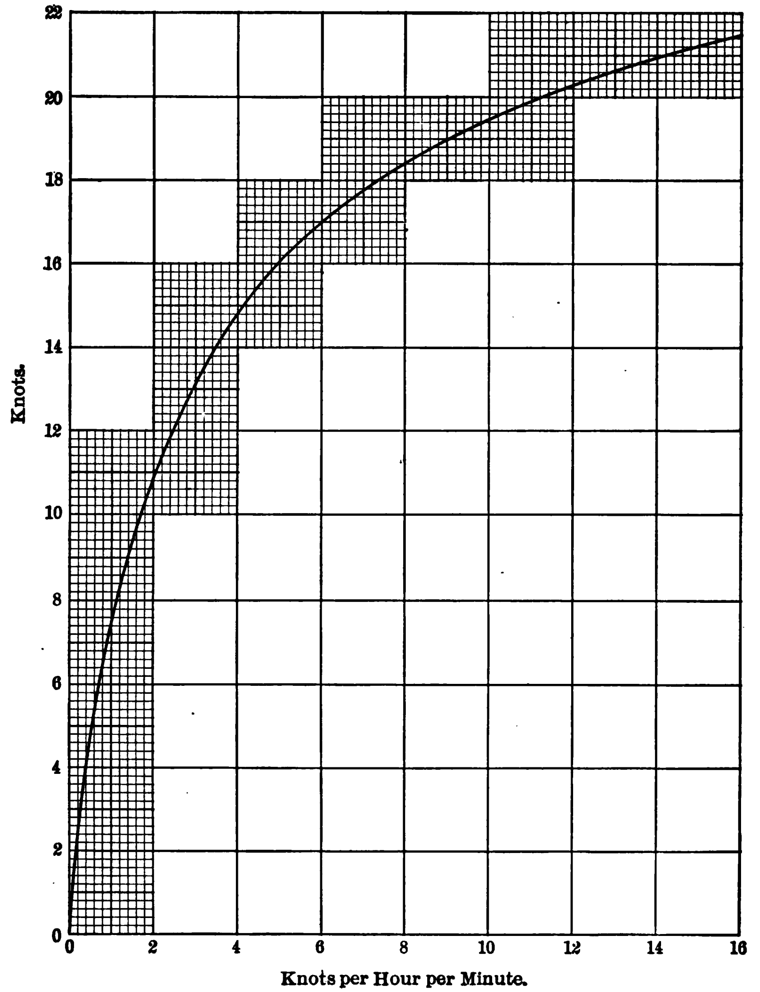


FIG. 36.

been plotted on Fig. 37. The actual retardation curve (obtained by experiment) is also shown in this figure. This curve can be represented very closely by an equation of the form—

$v(t+b)=a$. In this case the equation is of the form $v=\frac{52.92}{t+2.52}$, where v = speed of the ship and t = time intervals. It will be seen that the calculated, actual and equation curves all agree very closely. It will be noted that at the high speeds the change of speed is much more rapid than at the low speeds.

TABLE XI

"JUPITER," STOPPING

Knots.	E.H.P. (Bare Hull).	H.P. = 1.45 E.H.P.	a	Time to Decelerate 1 Knot.	Total Time from 21 Knots.
21	....	....	*14.0	*.0	*.0
20	....	....	*11.0	*.0800	*.08
19	11,800	17,100	9.0	.1000	.18
18	8,950	12,970	7.21	.1235	.3035
17	6,525	9,450	5.56	.1565	.4601
16	5,150	7,470	4.67	.1860	.6461
15	4,075	5,910	3.94	.2320	.8781
14	3,200	4,640	3.31	.2755	1.1536
13	2,500	3,630	2.79	.3280	1.4816
12	1,950	2,830	2.36	.3878	1.8694
11	1,500	2,175	1.98	.4560	2.3254
10	1,125	1,630	1.63	.556	2.8814
9	825	1,195	1.33	.676	3.5574
8	575	834	1.04	.840	4.3974
7	425	616	.88	1.042	5.4394
6	255	370	.616	1.338	6.7747
5	170	246.5	.493	1.8	8.5774
4	98	142	.355	2.36	10.9374
3	50	72.4	.242	3.34	14.2774
2	25	36.3	.181	4.94	19.2174
1	10	14.5	.145	6.14	25.3574
0	0	0	0	0	0

* Obtained by extending the curve.

Following the above method, the retardation curves for the U.S.S. *New Mexico* have been determined, and these will be used later on in the chapter when the subject of "backing" is treated.

The *New Mexico's* strut area (two on one side of the ship) = 13.3 sq.ft.

Area of two propeller hubs = 11.94 sq.ft.
Total strut area = 13.3 + 11.95 = 25.24 sq.ft.
Projected area of two propellers = 107.7 sq.ft.
Strut resistance = 9.4 per cent.

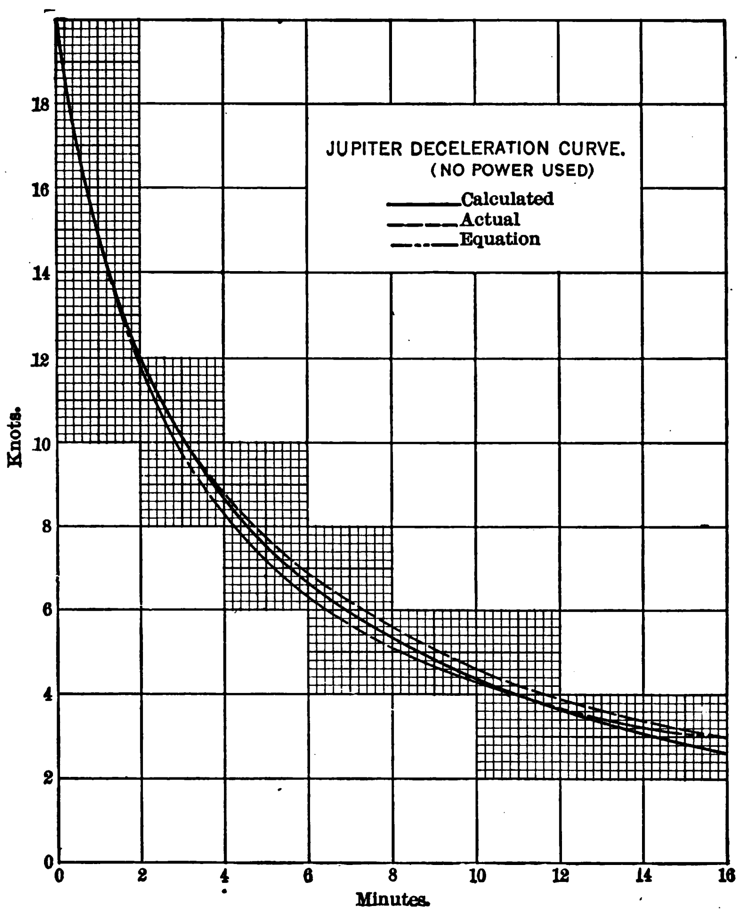


Fig. 37.

Propeller resistance $= 9.4 \times \frac{107.7}{25.24} = 40.2$ per cent.

Appendage resistance = 14.9 per cent.

Total added resistance (to bare hull) = 14.9 + 40.2 = 55.1 per cent.

Substituting $\Delta=32{,}000$ tons in the a equation previously reduced, we have

$$a=\frac{\text{H.P.}\times.005191}{v}.$$

Table XII

"NEW MEXICO," STOPPING

Knots.	E.H.P. (Bare Hull).	H.P. = E.H.P. × 1.551.	a
22	21,200	32,900	7.77
21	16,300	25,300	6.25
20	13,400	20,800	5.40
19	11,200	17,400	4.75
18	9,400	14,600	4.21
17	7,800	12,100	3.69
16	6,350	9,850	3.19
15	5,100	7,920	2.74
14	4,100	6,370	2.36
13	3,250	5,050	2.02
12	2.550	3,960	1.71
11	1.950	3,030	1.43
10	1,500	2,330	1.21
9	1,100	1,710	.986
8	775	1,205	.781

Table XII gives the calculations for a and the curve is plotted in Fig. 38. From this curve the knots retardation curve can be plotted as in the case of the *Jupiter*. The curve of r.p.m. to drive at these speeds can next be plotted, and taking 70 per cent of this as the dragging r.p.m. this curve also can be plotted. They are all shown in Fig. 38.

The equation for the knots retardation curve is

$$v=\frac{81.4}{t+3.87}.$$

The sudden drop in the r.p.m. when power is taken off, from 175 to about 122, corresponds to results obtained by experiment on the *Jupiter* and also to results obtained in the model tank.

This sudden large drop is a very material help to the induction motor when backing as it makes a larger torque available for the reversal of the screw.

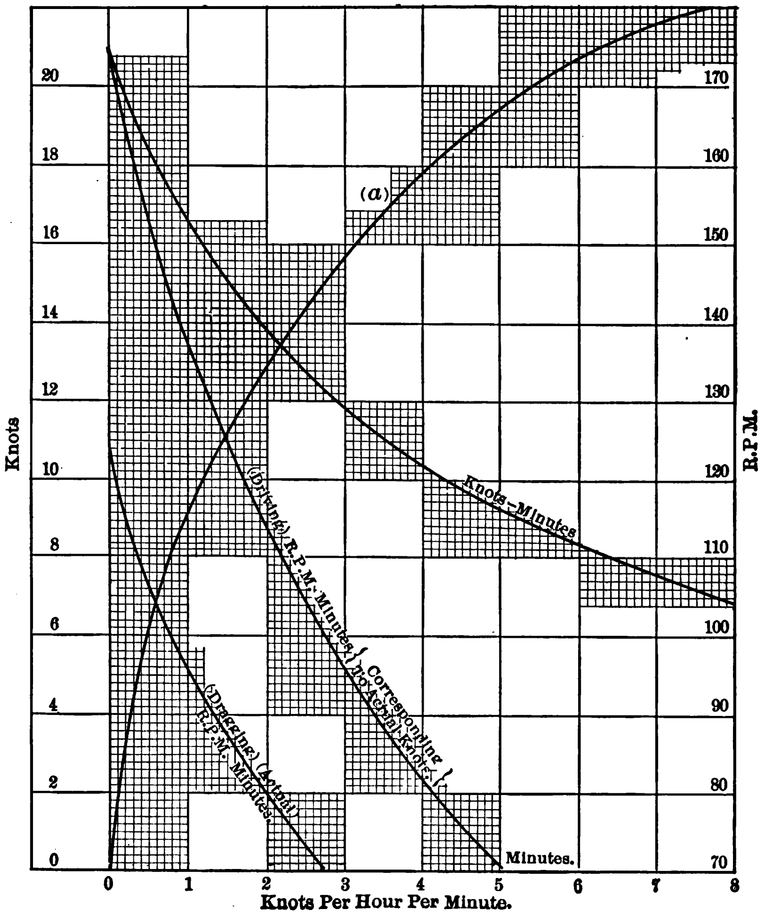


FIG. 38.

BACKING

The subject of "backing" seems to have been very little considered in the past. The main reason for this probably was that with reciprocating engines the backing power was ample

and was fixed by the design of the engine itself. When the marine turbine first entered the field of marine propulsion the subject of "backing" became a very live one. The first backing turbines built were wholly inadequate for the purpose, and this has resulted in more stringent requirements in this particular; but the term "backing power" is very vague and does not really define anything. In order to properly specify what the backing power of any ship should be, the speed at which the ship is moving through the water when she develops this power should be specified, as the latter limits the former, as will be seen later on. The limit of the power of any engine when backing is defined by the maximum attainable torque of the engine. This will be better understood after a study of Fig. 39. These curves were obtained by trials in the model tank, using a model of the U.S.S. *Delaware's* screws. The speeds of the ship are plotted both ahead (+) and astern (−) as well as the r.p.m. when turning ahead (+) and the r.p.m. when backing (−). From these curves the torque of the propeller can be taken off for any given condition of speed and r.p.m. These trials were run with the screw free of the model, so that it was running in undisturbed water, and consequently a wake factor will have to be applied to obtain actual ship conditions. In this case a factor of 14.5 per cent has been used, as this brings the actual torque of the ship, when driving ahead, into fairly close accord with the model results. For example, 122.2 r.p.m. corresponds to 21 knots speed; to find the speed on the curves corresponding to 21 knots, take $\frac{21}{1.145} = 18..35$ knots. The torque, from the curves corresponding to 122.2 r.p.m. and 18.35 knots is 465,000 lb.-ft. and the actual torque developed by the engine was 464,500 lb.-ft. Using this same method for all speeds Table XIII has been prepared, and this shows fairly close accord between actual engine torque and propeller torque. However, it is not intended to use this model-tank curve for actual values but only for comparative ones.

There are two very striking phenomena to be noticed about these curves. The first is that, with speed of ship constant,

CONTOUR CURVES OF TORQUE OF THE U.S.S. DELAWARE PROPELLER
EXPRESSED IN POUND FEET FOR ONE PROPELLER, DERIVED FROM EXPERIMENTS WITH MODEL PROPELLER NO. 285 AND BASED ON ASSUMPTION THAT LAW OF COMPARISON HOLDS FULLY AND THAT THERE IS NO CAVITATION.
DIAMETER PROPELLER = 18 FT. 3 INS. PITCH = 19 FT. 9 INS.

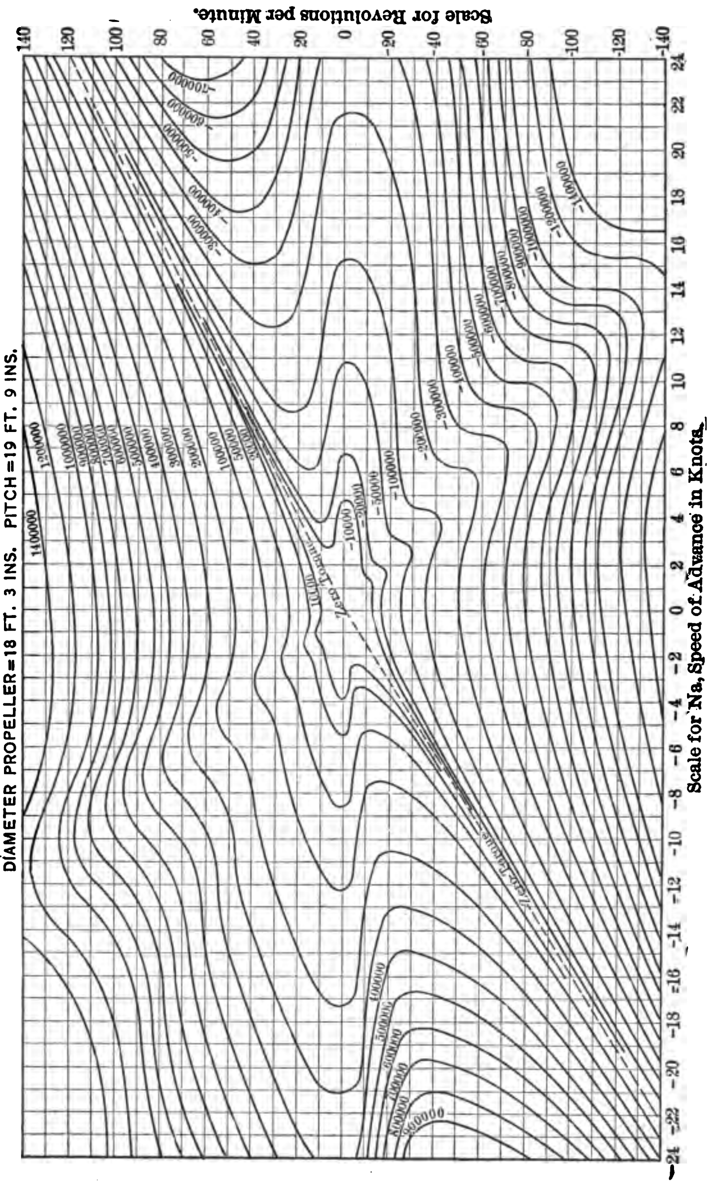


FIG. 39.

the torque of the propeller, as its revolutions per minute are reduced, passes through a high maximum torque before it reaches zero r.p.m. In other words, it requires a greater torque to bring the screw to rest than it does to hold it at rest. The second is that in backing, with constant r.p.m., the torque of the screw decreases as the ship slows down until a certain speed is reached, when the torque begins to increase; it reaches a maximum and then decreases again before the ship becomes stopped. Both of these phenomena have been verified by actual experiments on the *Jupiter*. They will each be taken up and considered in detail.

TABLE XIII

"DELAWARE"

Knots.	R.p.m.	I.H.P.	Actual Torque.	Curve Torque (14.5 Factor).
21	122.2	23,450	464,500	465,000
20	113.75	18,100	385,000	380 000
19	107.3	14,700	331,500	332,000
18	101.5	12,200	290,500	292,000
17	95.75	10,250	258,800	255,000
16	90.0	8,600	231,000	230,000
15	84.25	7,050	202,300	201,500
14	78.25	5,700	176,300	175,500
13	72.50	4,550	151,800	145,500
12	66.60	3,550	129,000	125,000
11	60.75	2,650	105,400	100,000
10	55.00	1,880	82,700	83,000

To illustrate the first point Fig. 40 has been plotted from the curves in Fig. 39. The ordinates of this curve represent per cent of the normal ahead driving torque and the abscissas represent per cent of ahead r.p.m. corresponding to the speed which it is assumed the ship is making. The ship is assumed to be making a constant speed ahead at all points represented on this curve. Starting at the right of the curve, it is seen that when power is taken off the engines, leaving the propellers free, the r.p.m. drop to about 76 per cent of the previous revolutions.

In the early part of this chapter it was shown, by experiment, that the *Jupiter* r.p.m. dropped to about 68 per cent. If reverse torque is now applied to reverse the screw and is gradually increased, the r.p.m. will gradually slow till a point is reached where the propellers are making 40 per cent of the ahead r.p.m.; at this point about 95 per cent of the ahead torque will be required; from this point on down to stop less torque will be required to slow the propeller; when the propeller becomes stopped the torque has reached a minimum, and will rise again

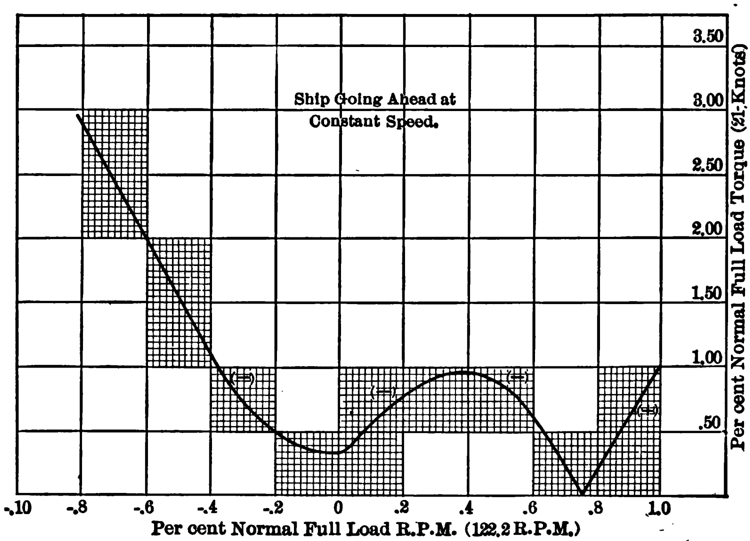


FIG. 40.

if the screw is actually reversed. The curve given was plotted for the condition of ship going ahead at 21 knots, but it is approximately correct for all speeds, as will be seen by following out the various speeds in Fig. 39. In Fig. 41 is given a similar torque curve for the *Jupiter*. This curve was determined by experiment in the following manner. With the ship going ahead at 14 knots power was suddenly thrown off; the propeller speed dropped to the point marked zero torque. The excitation of the generator was then reduced as much as possible and the backing switches thrown in; the propellers kept revolving ahead and the

excitation was gradually increased till the propellers just passed over the maximum torque point and started to reverse; the excitation was then reduced to just enough to keep the propeller stopped. There were two points for the stopped condition, one at which the propeller would just start revolving ahead and the other at which the propeller would just back; the curve has been run between the two points. At each point the elapsed time from the beginning of the experiment was taken and, by

PROPELLER TORQUE CURVE

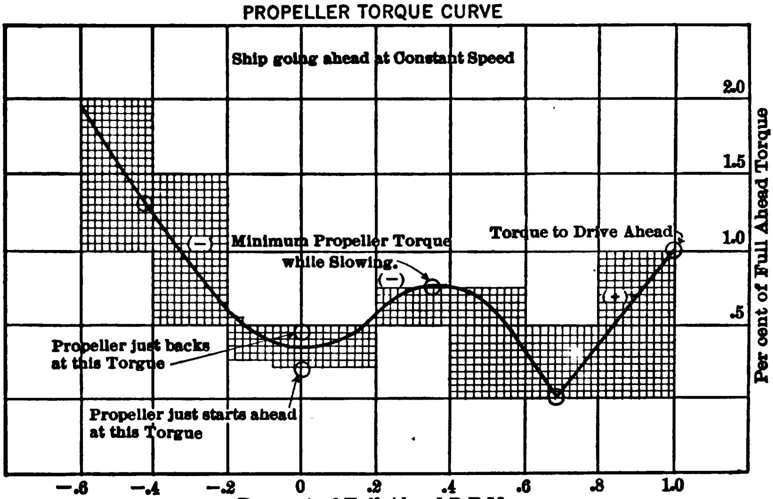


FIG. 41.

means of the retardation curves in Fig. 42, all points were reduced to the same speed. The curves in Fig. 42 were made up from " dragging " data taken at the time and from the retardation curve given in Fig. 35. The torques at the two points, that is the maximum and minimum points, were determined by the excitation at these points and were obtained from the torque curve of the motors. Fig. 43 shows this torque curve of the motor with 245 amperes excitation; the torque for the actual excitation used at the various points was assumed to vary as the square of the excitation. It will be seen that the curves in Figs. 40 and 41 are similar, but that the actual screws have a maximum

point (during reversing) that is lower than the model; the model shows a maximum point of 95 per cent while the *Jupiter's* maximum point is only about 75 per cent. Trials conducted on other model screws in the model tank showed this point to vary from 80 to 100 per cent. The data obtained from the *Jupiter* would indicate that these values are too high. However, in

REVOLUTION DECELERATION CURVE

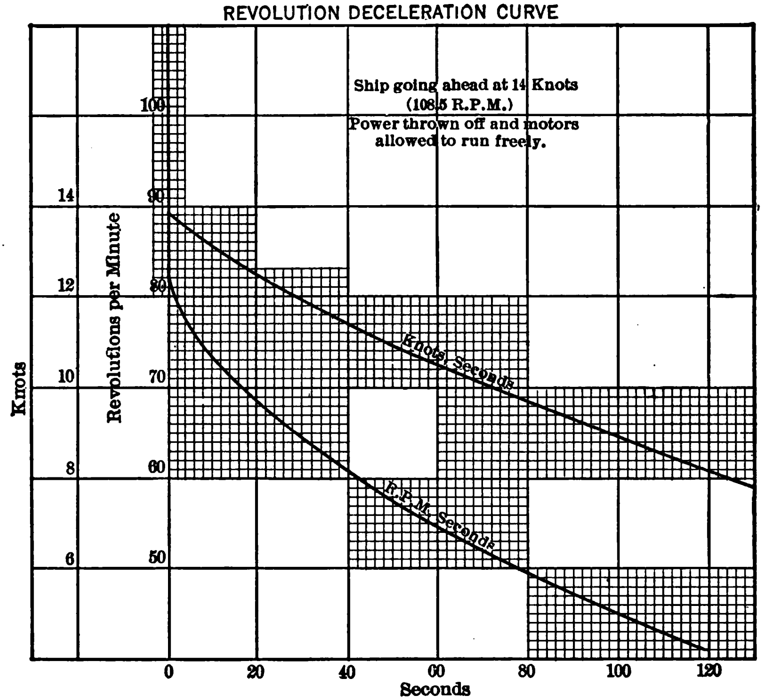


FIG. 42.

designing induction motors for backing it is not safe to have the torque on the "out of synchronism" part of the curve drop below 100 per cent of the ahead driving torque. That will insure a safe margin for getting past this " hump " in the torque curve.

The second phenomenon of the torque curves of Fig. 39 is illustrated in Fig. 44. This shows a ship backing with a constant number of r.p.m. from any given speed till the ship is stopped. The r.p.m. assumed are those which will give 100 per

cent ahead driving torque at the instant of backing. This curve is approximately correct for all speeds. From it it is seen that

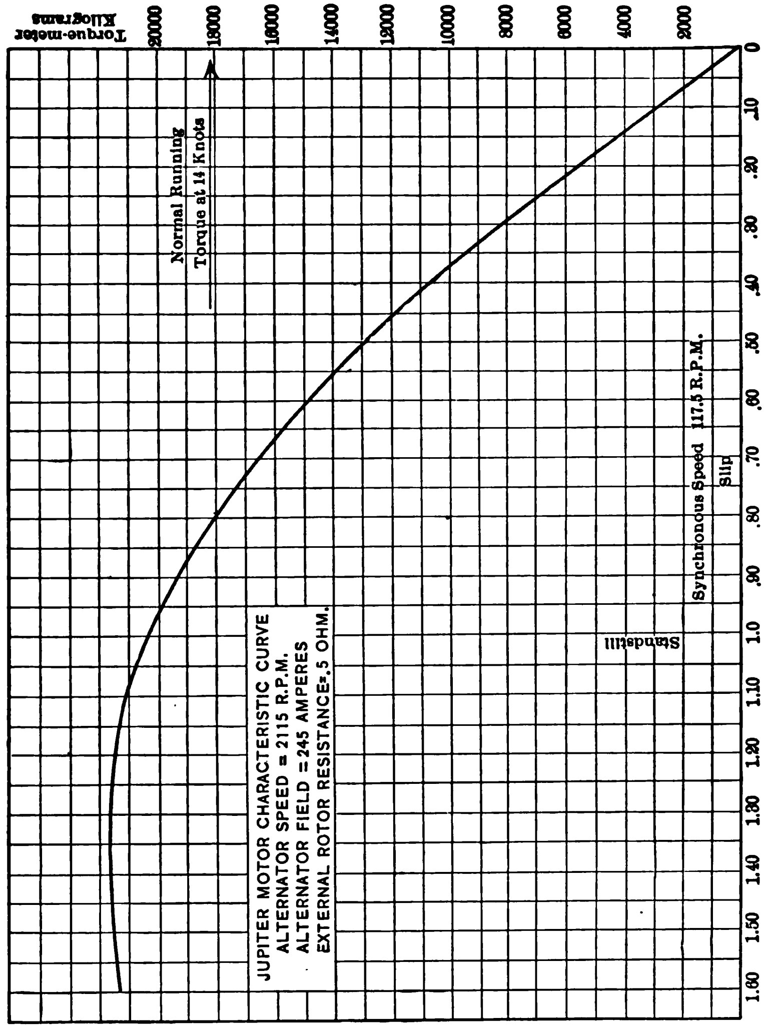


FIG. 43.

the torque necessary to turn the screw at the given r.p.m. falls as the ship slows till about 22 per cent speed is reached, when the torque begins to rise and continues to rise till 5 per cent speed

is reached, when it begins to fall again. This curve was also verified by experiment on the *Jupiter*. Fig. 45 shows two sets of backing trials conducted on the *Jupiter*. These were made with the ship going 14 knots and then suddenly reversing, using the resistances in the motors, and keeping the generator at a constant speed of 1950 r.p.m. and a constant excitation. Under these conditions the speed of the motors would be determined by the intersection of the propeller-torque curve of Fig. 41 and the motor-torque curve of Fig. 43. As the ship slowed the propeller-torque curve dropped lower so that the motors speeded up, but a maximum point was finally reached and the motor speed

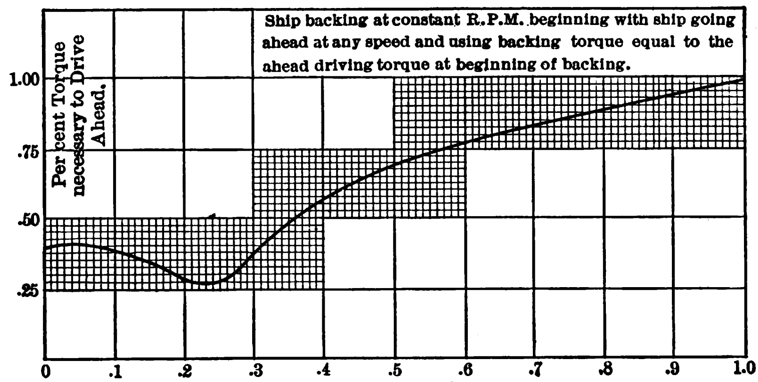


FIG. 44.

began to decrease, showing that the necessary driving torque had begun to go up. The motors dropped to a minimum and then speeded up again as the torque began to drop again. This follows the conditions of Fig. 44 exactly.

To explain this point further Figs. 55 and 56 are given. Fig. 55 is plotted from the model tank data given on Fig. 39, and Fig. 56 was obtained by actual experiment on the *Jupiter*. In these figures, as in Fig. 44, the ship is backing with constant revolutions till the ship is stopped; however, the revolutions chosen are the same as were used in going ahead, at the given speed, and this requires about three times as much torque as when going ahead. The actual speed of the *Jupiter* when

going ahead for this test was 39.5 r.p.m., or 5 knots, and the actual r.p.m. when backing was 39.5. The curve was obtained

BACKING AT 14 KNOTS.
Generator Speed Constant, 1950 R.P.M.
1st Trial Excitation 245 Amperes.
2nd Trial Excitation 225 Amperes.
Ship Stopped in About 4 Min. 45 Sec. in Each Case
Resistance Kept in All the Time.

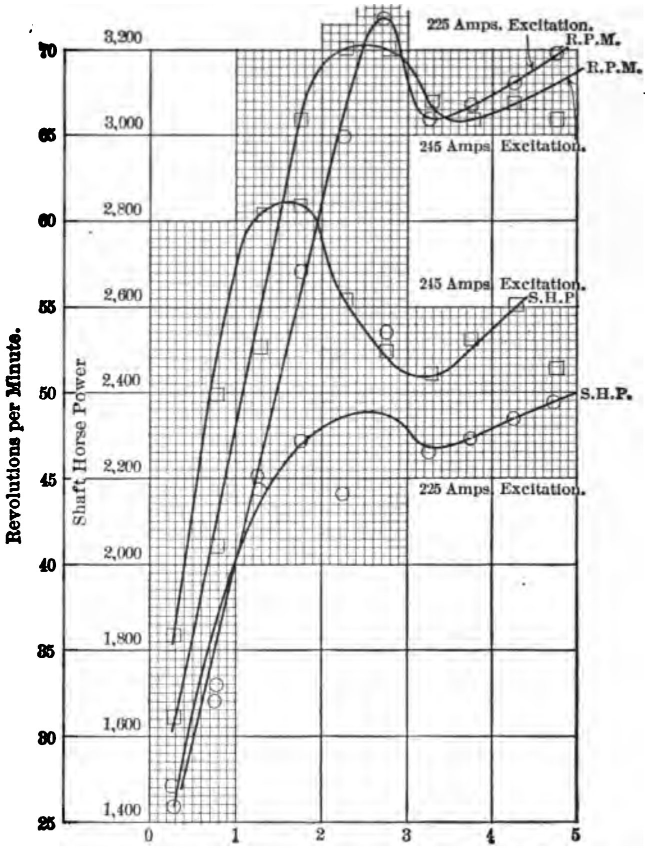


FIG. 45.

by taking H.P. and r.p.m. readings every five seconds and reducing the H.P. to torque. The curve obtained from the *Jupiter* is similar in all respects to that obtained from the *Delaware's*

model. The minimum torque comes at 65 per cent for the *Jupiter* and 60 per cent for the *Delaware;* the maximum point comes at 12 per cent for the *Jupiter* and 15 per cent for the *Delaware*. The maximum and minimum torques, however, are much lower in the case of the actual screw than in the case of the model.

From an inspection of Fig. 39 it will be seen that the conditions of (1) ship going ahead and propeller going from ahead to astern and (2) ship going astern and propeller going from astern to ahead are similar and should give similar torque curves when plotted. Also that the conditions of (1) ship backing at constant r.p.m. from any speed ahead to speed astern corresponding to the given r.p.m. and (2) ship going astern at any speed and propellers going ahead at a constant r.p.m. till ship is stopped and brought to speed ahead corresponding to the given r.p.m. are similar.

To illustrate these points Fig. 57 has been plotted. This curve or set of curves takes the screw through the entire cycle of conditions. It is made up of the various curves that have already been considered. Starting with the ship going ahead, the screw is suddenly reversed and brought up to revolutions which will give the same torque as was used when going ahead; holding these revolutions constant, the ship is backed till she stops; this is further continued till the ship picks up speed astern corresponding to the revolutions; the screw is then suddenly reversed and revolutions brought up to those which the problem started with (the latter part of this curve is taken beyond practical limits of actual screws as it runs the torque up too high, but it was chosen so as to make a complete cycle and end up at the starting point); these revolutions are maintained till the ship stops; they are continued further till the ship picks up speed corresponding to these revolutions, which brings conditions back to the starting point. Part 1 shows the cycle through which the torque passes while the screw is being reversed. Part 2 shows the change in torque while the ship is slowing down, backing at constant r.p.m. Part 3 shows the ship picking up sternboard with screw going at same r.p.m. Part 4 shows the

torque cycle of the screw when it is suddenly reversed to go ahead, ship still going astern. Part 5 shows the torque cycle while the ship is slowing down, propeller going ahead at constant r.p.m. Part 6 shows the torque cycle of the propeller while ship is picking up speed ahead, propeller turning ahead at same revolutions as before. From inspection it will be seen that parts 1 and 4 are similar curves, parts 2 and 5 are similar and parts 3 and 6 are similar. Part 5 was taken so far over on the chart that it does not show the drop in the torque before the rise comes, in other words, it is part 2 beginning to the left of the minimum point. Parts 1, 2, and 3 have already been verified by test on the actual screws of the *Jupiter*, and to make the verification complete Fig. 58 is given. The curves obtained here were obtained under the same conditions as those in Fig. 45, that is, the generator was kept at constant speed and excitation and the motors were run with resistances in; under these conditions the speed of the motors would be determined by the intersection of the propeller torque and the motor-torque curve given in Fig. 43. The *Jupiter* is carried through the same cycle in Fig. 58 that the model screw is in Fig. 57, that is, the ship was going ahead 12 knots and the screws suddenly reversed; the ship then backed till she had full sternboard; the screws were again reversed and kept going ahead till the ship had full ahead speed. The points where the ship stopped are noted. From an inspection of Fig. 43 it will be seen that on the right-hand side the torque curve of the motor is practically a straight line, so that revolutions vary directly as the torque. The curves plotted in Fig. 58 are revolutions of the screw, but they may also be taken as torque on the motor shaft simply by reversing the curves, that is when r.p.m. are increasing torque is decreasing, when r.p.m. reaches a maximum torque reaches a minimum, and so on. It will be seen that the curves are similar to parts 2, 3, 5 and 6 of Fig. 57; also the two parts of Fig. 58 are similar. This confirms the correctness of the shape of all the curves given in Fig. 57.

Now that it has been shown how a propeller acts during the entire cycle of backing, from the instant the power is removed

till the ship is stopped, some cases of actual backing will be taken up. As previously stated, the power any engine is capable of delivering while backing is limited by the maximum torque of the engine. To make this plainer the engines of the *Delaware* are taken as an example, and two theoretical indicator cards have been constructed and are shown in Fig. 46. The heavy-line curve shows the card when going at full power, the data for the card being taken from the full-power trial. The dotted card shows the conditions if full boiler pressure could be obtained

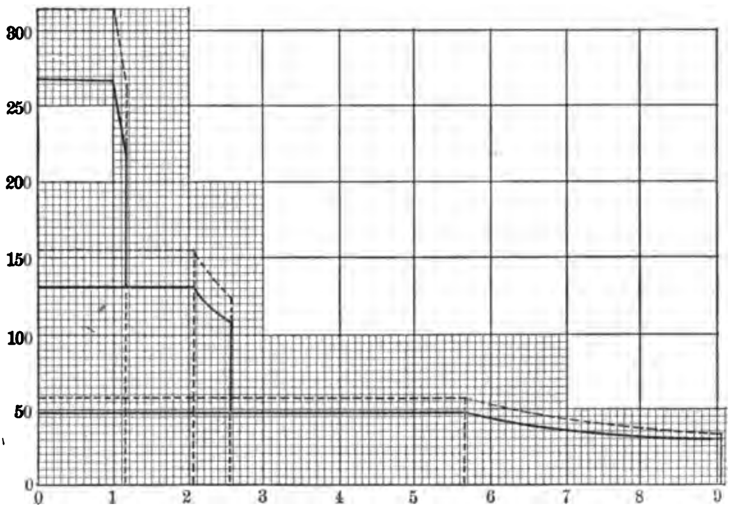


FIG. 46.

in the high-pressure valve chest. In the construction of these cards no account has been taken of wire-drawing or clearances, as they are only for the purpose of illustration. The data used in the construction of the cards are as follows:

Diameter H.P. cyl.	$=38.5$ in.
Diameter I.P. cyl.	$=57.0$ in.
Diameter L.P. cyl. (2)	$=76.0$ in.
Stroke	$=48$ in.
H.P. cut off	$=.86$ in.
I.P. cut off	$=.8$ in.
L.P cut off	$=.62$ in.

For heavy-line card, pressure in high-pressure valve chest = 268 lb. absolute. For dotted-line card, pressure in high-pressure valve chest = 315 lb. absolute. Back pressure = 5 lb. absolute in both cases.

The area of each card represents work per stroke, and since the stroke is the same for each card, the areas can also be used to represent torque. The area of the dotted curve is 19 per cent greater than the full-line curve; that is to say, the engines could develop only 19 per cent more torque than the ahead full-power torque if full boiler pressure could be obtained in the high-pressure valve chest. Actually, on backing trials, the highest torque attained was about 9 per cent greater than the torque developed on the full-power ahead trial. By reference to Figs. 40 and 41 it will be seen that this torque is reached when backing at about 40 per cent of the revolutions necessary to drive ahead, if the ship is going ahead full speed and the engines are backing. This means that about 43.6 per cent (1.09×40) of full ahead power will be developed when the ship first begins to back. This amount will, of course, be increased as the ship slows. In the case of turbine ships the torque is far less than in the case of reciprocating engines, probably not more than half, so that they probably do not develop more than one-quarter of full ahead power at beginning of their backing. An induction motor can be designed to give a much greater maximum torque than the normal driving torque; also, since induction motors for ship propulsion will have two sets of pole connections, the motors can be arranged to back on the slow-speed connection. This will allow the turbine to run at nearly full speed while backing at a low number of propeller r.p.m. This condition is ideal for getting high power while backing; the motor is capable of producing large torque and the turbine is running at a sufficiently high speed to enable it to develop full power. In other words, the turbine condition when backing on the slow-speed connection is practically the same as when going ahead at full power on the high-speed connection. This should give very fine backing results.

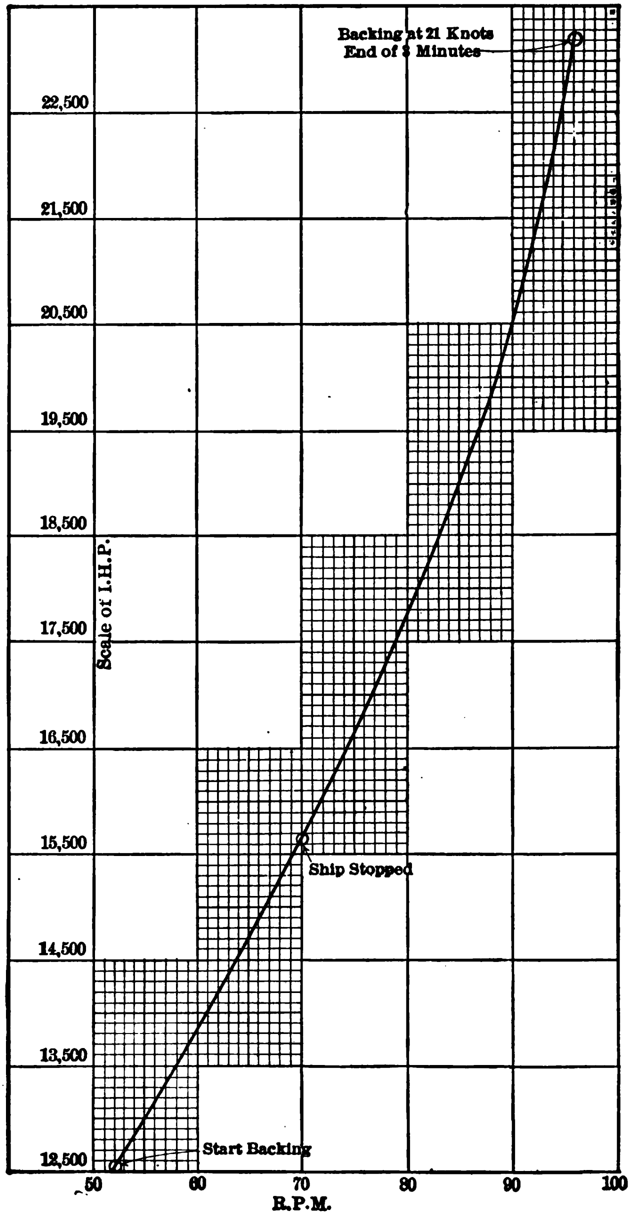

Backing at 21 Knots
End of 3 Minutes
22,500
21,500
20,500
19,500
18,500
Scale of I.H.P.
17,500
16,500
15,500
Ship Stopped
14,500
13,500
12,500
Start Backing
50
60
70
80
90
100
R.P.M.

FIG. 47.

Table XIV

"DELAWARE," BACKING

Knots.	I.H.P.	I.H.P. × P.C.	E.H.P. (all app.)	Total H.P.	a	Interval Time in Minutes.	Total Time.
21	....	....	....	....	....	....	....
19	12,500	8,170	9,900	18,070	7.9	.2925	.2925
18	12,665	8,275	8,500	16,775	7.75	.1276	.4201
17	12,825	8,400	7,225	15,625	7.65	.1299	.5500
16	12,990	8,500	6,000	14,500	7.53	.1315	.6815
15	13,150	8,600	4,900	13,500	7.48	.1331	.8146
14	13,315	8,720	3,900	12,620	7.49	.1335	.9481
13	13,475	8,820	3,100	11,920	7.62	.1322	1.0803
12	13,640	8,920	2,430	11,350	7.85	.1292	1.2095
11	13,800	9,030	1,880	10,910	8.24	.1242	1.3337
10	13,965	9,130	1,400	10,530	8.75	.1176	1.4513
9	14,125	9,250	1,050	10,300	9.50	.1095	1.5608
8	14,290	9,350	750	10,100	10.50	.1000	1.6608
7	14,450	9,450	500	9,950	11.82	.0897	1.7505
6	14,615	9,560	300	9,860	13.65	.0784	1.9289
5	14,775	9,660	190	9,850	16.35	.0667	1.8956
4	14,940	9,770	120	9,890	20.55	.0542	1.9498
3	15,100	9,880	60	9,940	27.50	.0416	1.9914
2	15,265	9,980	20	10,000	41.50	.0290	2.0204
1	15,430	10,100	10	10,110	84.00	.0160	2.0364
0	15,600	10,200	0	10,200	0	.0076	2.0440

To show what this means a comparison has been made with the *Delaware* when going ahead at 21 knots and the engines were suddenly reversed. Fig. 47 gives the data obtained on this trial. At the beginning she developed 12,500 I.H.P., which is 43.8 per cent of her ahead full power, and at stop she developed 15,600 I.H.P., which is 54.6 per cent of her full power. The *Delaware's* displacement is 20,000 tons. Substituting this value of Δ in the a equation, there results,

$$a=\frac{\text{H.P.}\times .00831}{v}.$$

Substituting the data given by Fig. 47 in this equation, Table XIV has been calculated for the backing condition. The values of a obtained have been plotted on Fig. 48.

From the values of *a* given on this curve a retardation curve has been plotted on Fig. 49. This curve shows the total time to stop the ship to be two minutes three seconds. The actual

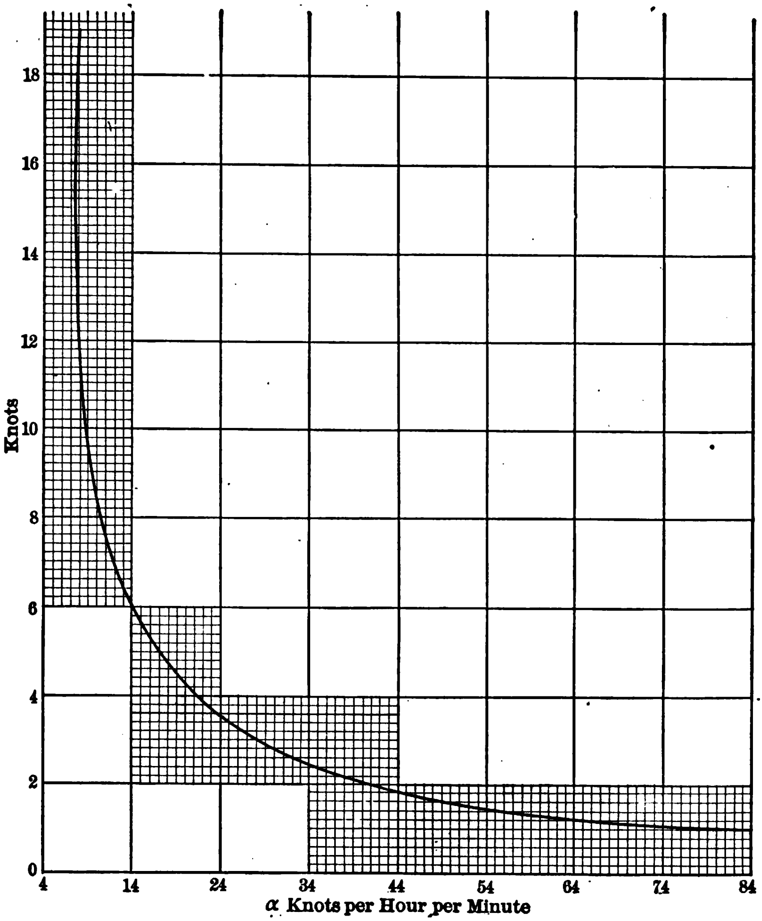


FIG. 48.

time as measured was two minutes twenty-one seconds. The results are considered to be very close, as it is difficult to determine the exact instant a ship becomes dead in the water. The same method of calculation has been followed for the *New*

Mexico. In her case $\Delta = 32{,}000$, so

$$a = \frac{\text{H.P.} \times .00591}{v}.$$

Table XV shows the results of the calculations for a. It has been assumed that full power is developed all the way through

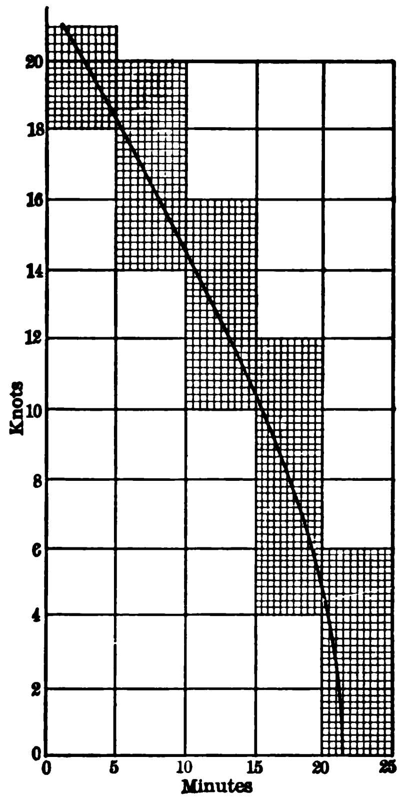


FIG. 49.

due to the large maximum torque of the induction motor. It has also been assumed that the ship drops in speed to 20 knots during the act of reversal. The values of a have been plotted on Fig. 50 and from these values the retardation curve has been plotted on Fig. 51. This curve shows the time necessary to

bring the ship to a stop to be 1 minute 50.4 seconds. It is realized, of course, that this condition may not be entirely

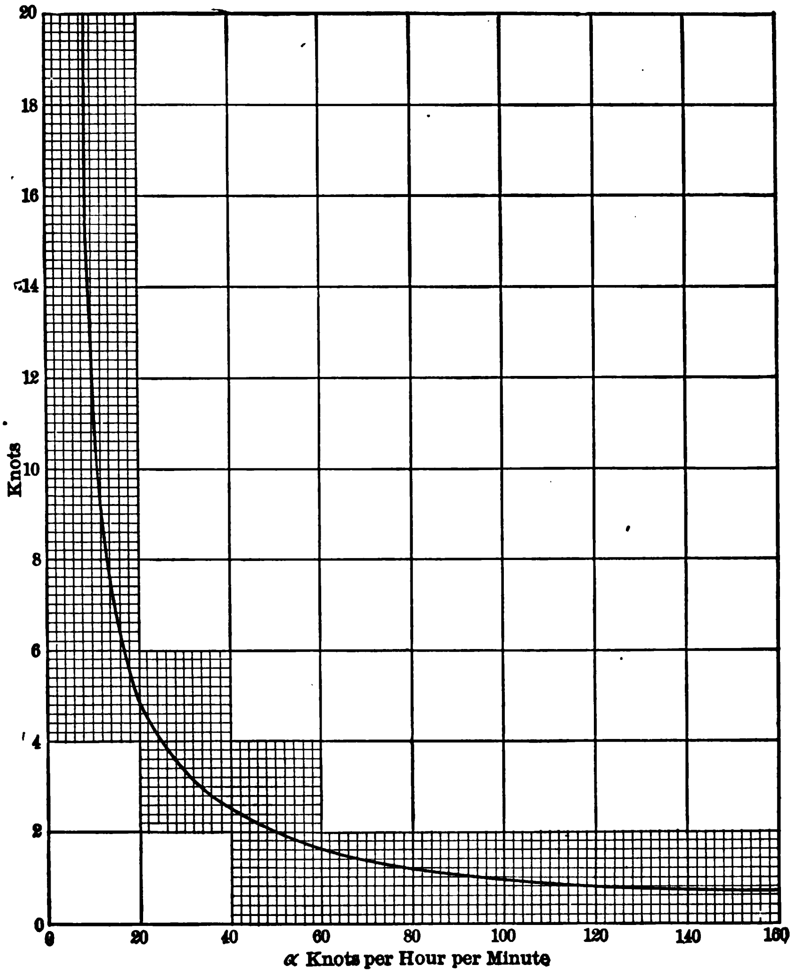


FIG. 50.

reached, as some reason (such as propeller vibration) which has nothing to do with the engines may make it undesirable to use this much power in backing.

TABLE XV

"NEW MEXICO," BACKING

Knots.	E.H.P. (Bare Hull)	E.H.P. (all) 14.9 per Cent. App.	S.H.P. × P.C.	Total H.P.	a	Mins. (Interval).	Time in Mins.
21	16,300	18,740	0,000	....	....	0	0
20	13,400	15,400	18,740	34,140	8.87	.232	.232
19	11,200	12,880	18,740	31,620	8.65	.114	.346
18	9,400	10,800	18,740	29,540	8.53	.1165	.4625
17	7,800	8,960	18,740	27,700	8.45	.1178	.5803
16	6,350	7,300	18,740	26,040	8.45	.1182	.6985
15	5,100	5,860	18,740	24,600	8.52	.1178	.8163
14	4,100	4,710	18,740	23,450	8.70	.1162	.9325
13	3,250	3,730	18,740	22,470	8.97	.1132	1.0457
12	2,550	2,930	18,740	21,670	9.38	.109	1.1547
11	1,950	2,240	18,740	20,980	9.90	.1036	1.2583
10	1,500	1,720	18,740	20,460	10.62	.0975	1.3558
9	1,100	1,260	18,740	20,000	11.55	.0843	1.4401
8	775	890	18,740	19,630	12.75	.0823	1.5224
7	525	600	18,740	19,340	14.35	.0738	1.5962
6	325	375	18,740	19,115	16.55	.0647	1.6609
5	175	200	18,740	18,940	19.70	.0551	1.7160
4	75	86	18,740	18,826	24.40	.0453	1.7613
3	50	58	18,740	18,798	32.55	.0351	1.7964
2	25	29	18,740	18,769	48.75	.0246	1.8210
1	10	12	18,740	18,752	97.50	.0137	1.8347
0	0	0	18,740	18,740	0	.00530	1.8400

TURNING

It has always been known that when a ship turns, the inboard screw slows down if the throttle is not touched during the turn, as is the rule ordinarily followed. In the case of an electrically propelled twin-screw ship, operating both propellers with one governor-controlled turbine, the r.p.m. of the two screws are maintained the same as they were before the turn. In order to determine exactly what effect this would produce, turning trials were carried out on the *Delaware* and the *Jupiter*. Six turns of 360° were made on the *Delaware*, two at 10 knots with 16° right rudder, two at 12 knots with 16° right rudder, two at 12 knots with 27° rudder. The first four turns are shown as curves in

Fig. 51. The data obtained on the last two turns are given in Table XVI. The data on tactical diameter, etc., for all six turns are given in Table XVII. It will be seen from Table XVII

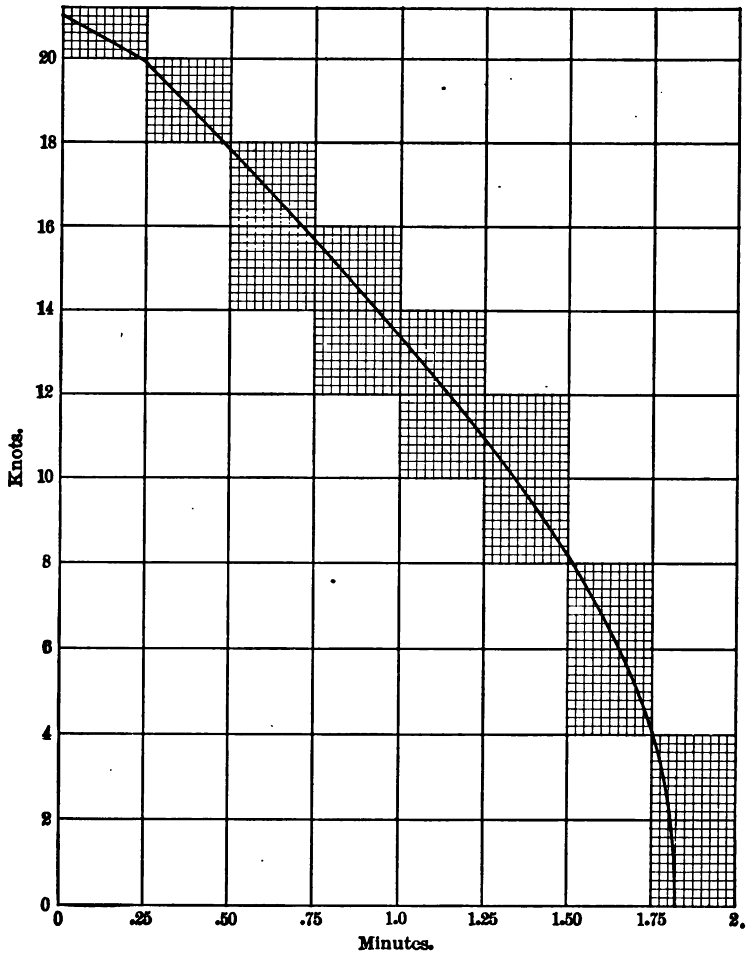


FIG. 51.

that, for speeds above 10 knots the tactical diameter is practically the same, whether the turn is made with r.p.m. constant or whether the inboard screw is allowed to slow down. In the turn at 12 knots, with 16° rudder and r.p.m. constant, it will be

seen that the I.H.P. of the inboard screw rose steadily as the turn progressed, and that the I.H.P. of the outboard screw first dropped and then rose steadily as the turn progressed. This latter peculiarity is not observed in the curves given on the 10-knot turn, probably on account of inability to indicate power frequently enough, but it is present in all of the turns made by the *Jupiter* as will be seen later on. The reason for the shape of the horse-power curves when turning is that the inboard screw is maintaining a constant r.p.m. at a much lower virtual speed of the ship than these r.p.m. would give if driving ahead, so, of course, the power goes up; the condition is still more exaggerated as the ship slows during the turn. The outboard screw, at the beginning of the turn, is maintaining a constant r.p.m. at a considerably higher virtual speed of the ship than these r.p.m. would give if driving ahead, so the power required drops; but as the ship slows during the turn a point is soon reached where this excess speed is lost, and as the speed of the ship falls lower the power commences to rise. From Table XVI, on the turn at 12 knots with 27° rudder, the greatest increase of power on the inboard screw was 73.5 per cent of the normal driving power; the greatest increase on the outboard screw was 4.2 per cent of the normal driving power, and the total increase of power was 39 per cent of the total driving power.

TABLE XVI

U. S. S. "DELAWARE," FEBRUARY 12, 1914.—TURNING TRIALS

Turn at 12 knots speed. 27° helm. Throttle untouched. Run No. 5.

I.H.P.	Pressures.			Revolutions.
	H.P. Valve Chest.	1st Receiver.	2d Receiver.	
Starboard, 1889	76	30	3	54
Port, 2235	75	24	1	66
Total, 4124				

Turn at 12 knots speed. 27° helm. Maintaining same revolutions during turn. Run No. 6.

I.H.P.	Pressures. H.P. Valve Chest.	Pressures. 1st Receiver.	Pressures. 2d Receiver.	Revolutions.
Starboard, 3469	170	60	10	66
Port, 2047	75	22	−1	66
Total, 5516				

Horse-power on straight run. From former records.

10 knots speed ...	Starboard engine	1175 I.H.P.	55 turns
	Port engine	1150 I.H.P.	
12 knots speed ...	Starboard engine	2000 I.H.P.	66 turns
	Port engine	1965 I H.P.	

In Figs. 52 and 53 are shown the results of turning trials on the *Jupiter*. Two turns were made at 12 knots and two at 14 knots; one turn was made to starboard and one to port in each case; the turn was made through 180° in each case using 25° rudder. It was possible to get very accurate results on these trials as the r.p.m. were maintained exactly constant by the governor, and horse-power readings were taken every five seconds. The curves obtained are all similar to those shown on the *Delaware's* 12-knot turn in Fig. 52. The reason for the shape of these curves has already been given under the explanation of the *Delaware's* curves. The difference in the curves obtained when turning to starboard and port are due, partly to the fact that the rudder angles were probably not exactly the same, and partly to the difference in the effect of wind and sea on the two sides. The greatest increase of power occurred on the 14-knot turn to port. The inboard screw increased in power 53.5 per cent, the outboard screw increased 4.5 per cent, and the total increase of power was 29 per cent. These percentages are considerably lower than those obtained on the *Delaware*, but the

TABLE XVII

TACTICAL DATA, U. S. S. "DELAWARE"

DRAUGHT, FOWARD, 28′ 8″; AFT, 29′ 11″

Date: February 12, 1914. Place: Off Ceiba Bank, Cuba. Depth of Water: 100 Fathoms

Serial number of run.	On Straight Course.			Rudder.			Engine Ahead.		90 Degrees.					180 Degrees.				360 Degrees.		Final speed on turn.	Final drift angle measured at foremast.
	Speed.	R.P.M. Starboard screw.	R.P.M. Starboard.	Direction.	Angle.	Time to put over.	Starboard.	Port.	Advance.	Transfer.	Time for turning.	Rev. Starboard.	Rev. Port.	Tactical diameter.	Time for turning.	Rev. Starboard.	Rev. Port.	Final diameter.	Time for turning.		
1	10	55	55	Right	16°	6″	Throttle untouched		700	425	3′ 07″	50	55	890	5′ 50″	48	55	890	11′ 43″	7.2	4½°
2	10	55	55	do.	16	6	Revs. of both maintained		750	495	2 19	55	55	1010	6 04	55	55	1010	11 58	7.8	1⅓
3	12	66	66	do.	16	6	Throttle untouched		750	460	3 37	60	66	940	4 57	59	66	940	9 57	9.0	4⅓
4	12	66	66	do.	16	6	Revs. of both maintained		810	490	2 54	66	66	980	5 10	66	66	980	10 04	9.75	2
5	12	66	66	do.	27	6	Throttle untouched		555	350	2 08	No	data	710	4 01	54	66	710	8 26	7.2	5
6	12	66	66	do.	27	6	Revs. of both maintained		600	340	2 00	"	"	710	3 53	66	66	710	7 58	8.9	4⅓

* Notes on wind and sea.—Wind E. N. E., force 1; sea, smooth.

Delaware's rudder is 50 per cent larger than that of the *Jupiter*, so that she turns and slows faster than the *Jupiter* and consequently takes a larger increase of power.

U.S.S. DELAWARE.

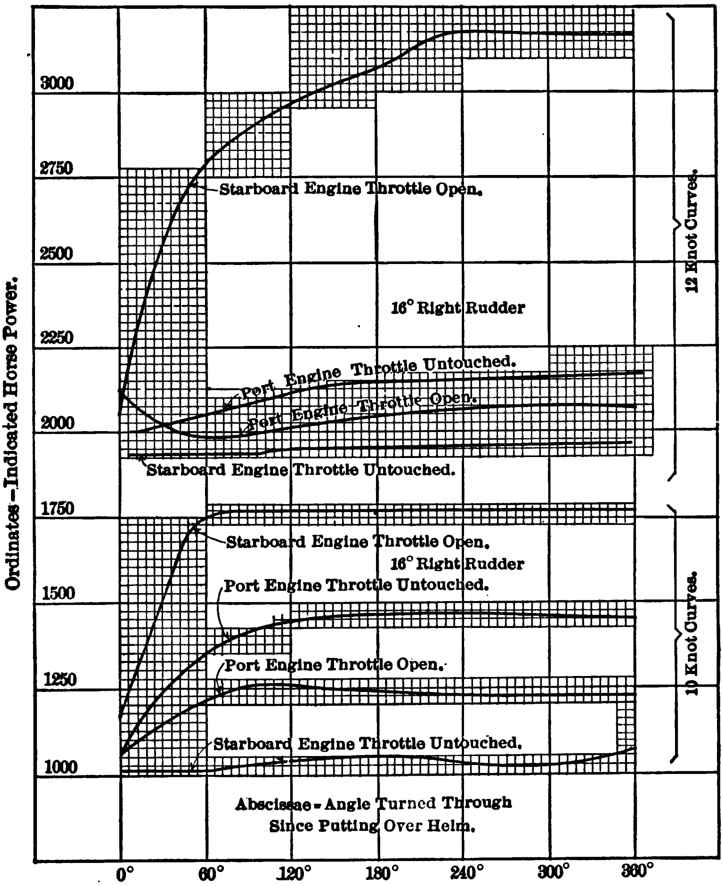


FIG. 52.

The effect of turning on an electrically propelled ship would depend on the design of the motors and the turbines. In the case of the *Jupiter* the power plant of the ship is sufficient to maintain constant r.p.m. at all speeds during a turn; also the maximum torque of the motors is sufficient to insure that they

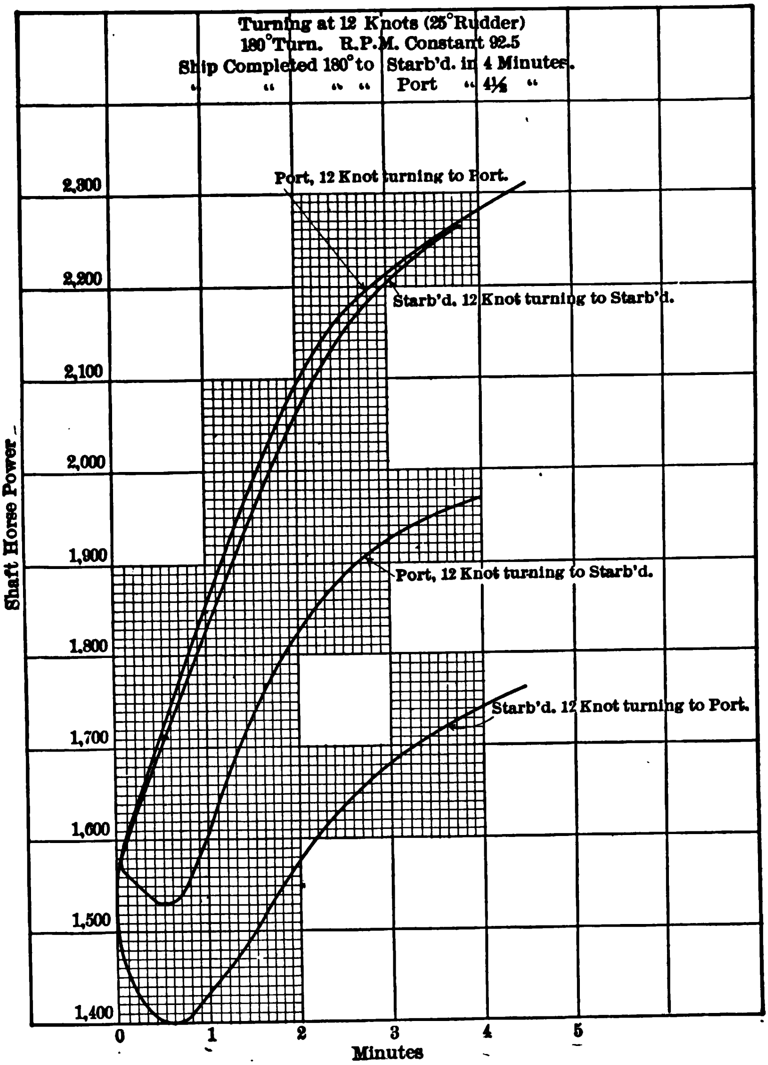

Turning at 12 Knots (25°Rudder)
180°Turn. R.P.M. Constant 92.5
Ship Completed 180° to Starb'd. in 4 Minutes.
" " " " Port " 4⅛ "
Port, 12 Knot turning to Port.
Starb'd. 12 Knot turning to Starb'd.
Port, 12 Knot turning to Starb'd.
Starb'd. 12 Knot turning to Port.
Shaft Horse Power
2,300
2,200
2,100
2,000
1,900
1,800
1,700
1,600
1,500
1,400
0
1
2
3
4
5
Minutes

FIG 53.

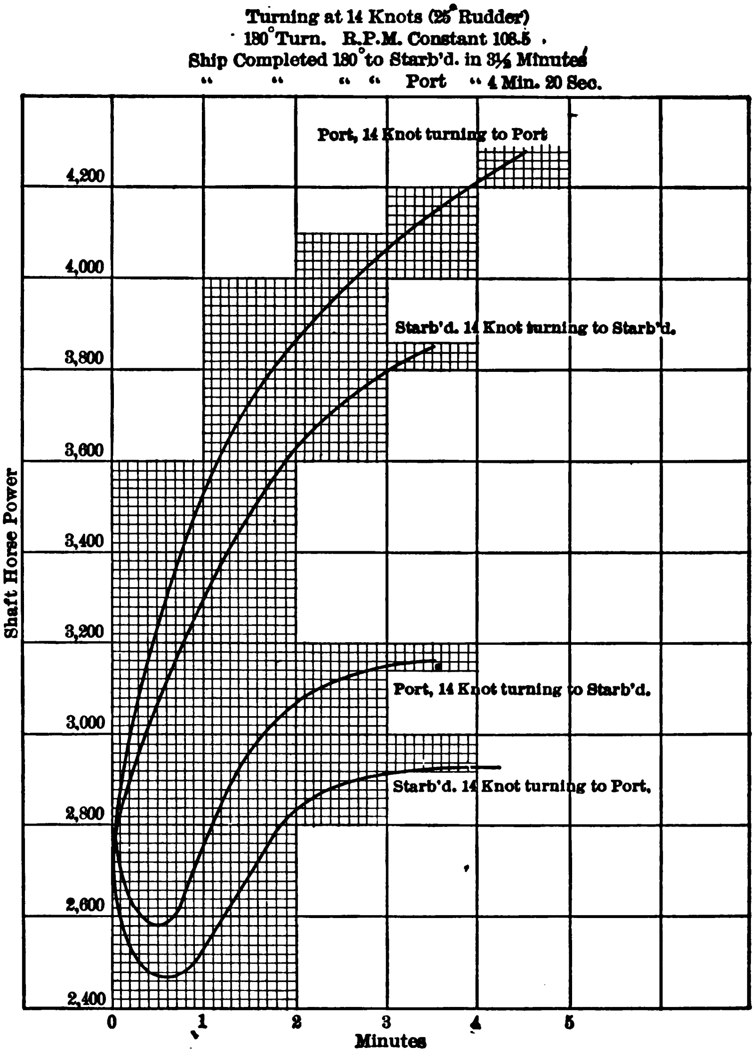

Turning at 14 Knots (25° Rudder)
180° Turn. R.P.M. Constant 108.5
Ship Completed 180° to Starb'd. in 3½ Minutes
" " " " Port " 4 Min. 20 Sec.
Port, 14 Knot turning to Port
Starb'd. 14 Knot turning to Starb'd.
Port, 14 Knot turning to Starb'd.
Starb'd. 14 Knot turning to Port.
Shaft Horse Power
4,200
4,000
3,800
3,600
3,400
3,200
3,000
2,800
2,600
2,400
0
1
2
3
4
5
Minutes

FIG. 54.

will stay in step during a turn if the proper excitation is maintained. In the case of the *New Mexico*, however, when turning at high speeds, either at 18 knots with one turbine or at 21 knots with two turbines, the boilers and turbines are not capable of giving a 39 per cent overload, and consequently the turbines will

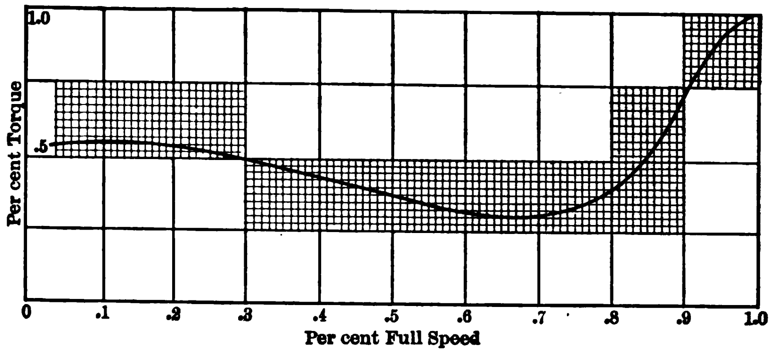


FIG. 55.

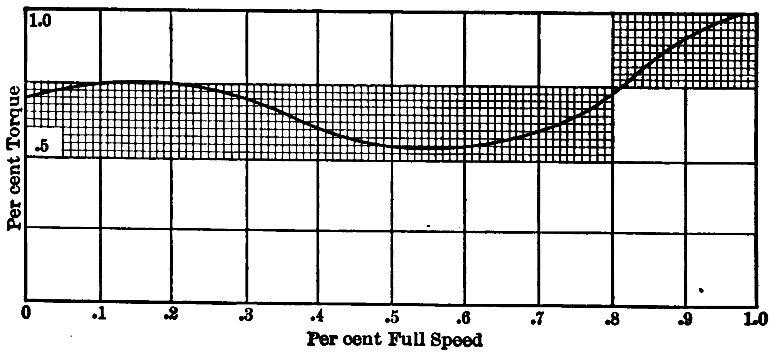


FIG. 56

simply slow during the turn, provided that the maximum torque of the motors is greater than that of the turbine. In other words, the *New Mexico* will turn without any reduction of r.p.m. at all low speeds, say below 16 knots, and will slow her r.p.m. during the turn at all speeds above this.

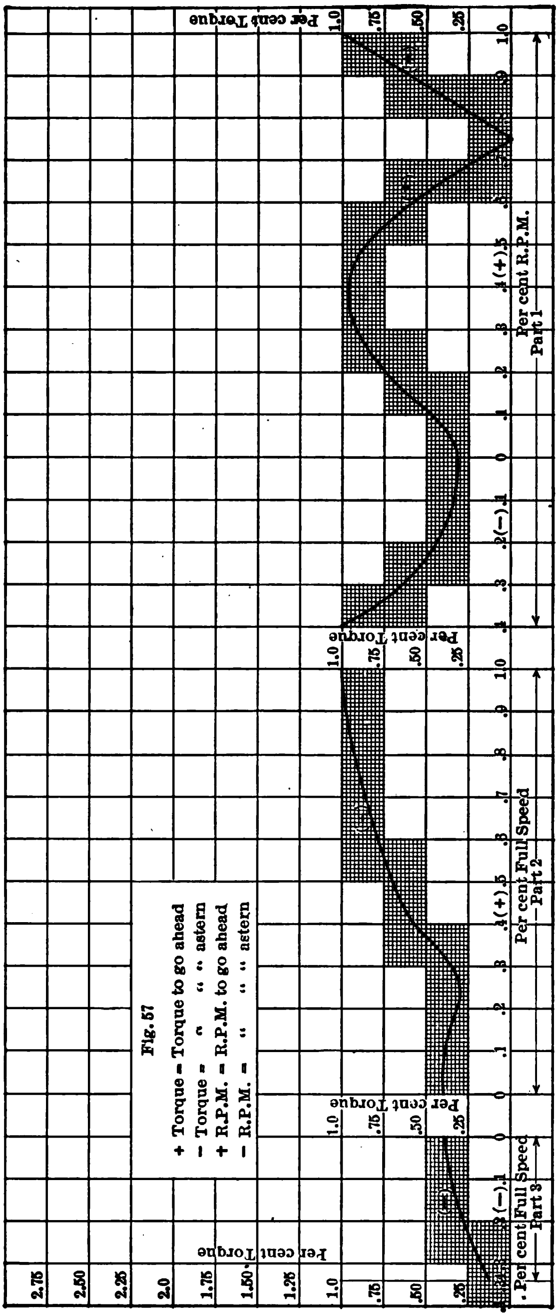


Fig. 57
+ Torque = Torque to go ahead
− Torque = " " " astern
+ R.P.M. = R.P.M. to go ahead
− R.P.M. = " " " astern
Per cent Torque
Per cent R.P.M.
Part 1
Per cent Full Speed
Part 2
Per cent Full Speed
Part 3

FIG. 57.

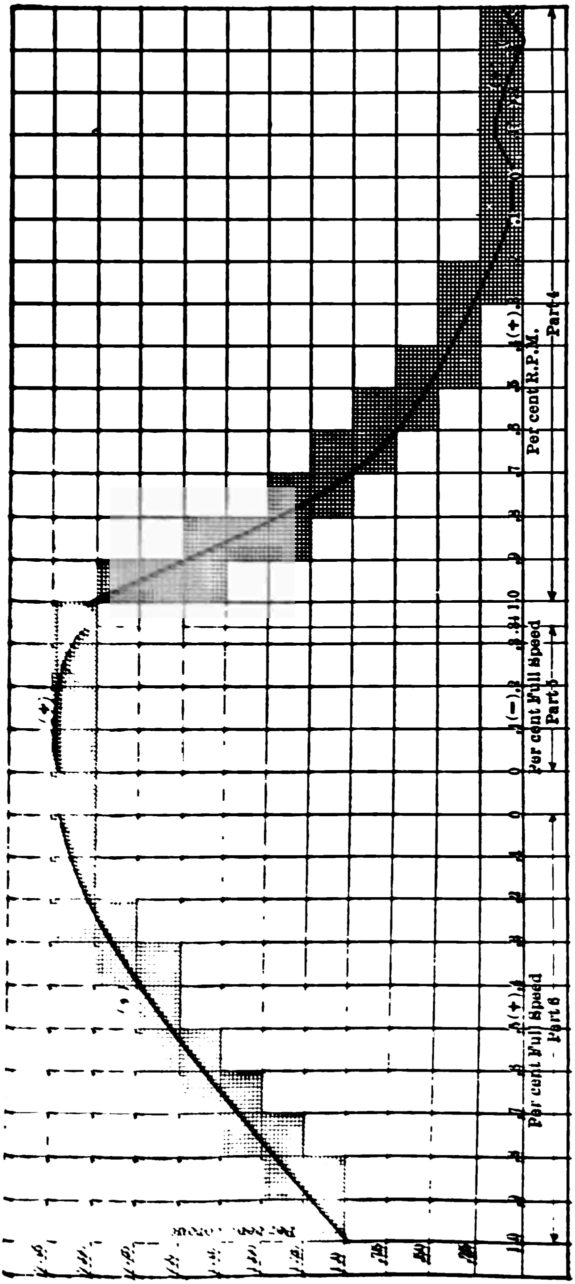


FIG. 57.—*Continued.*

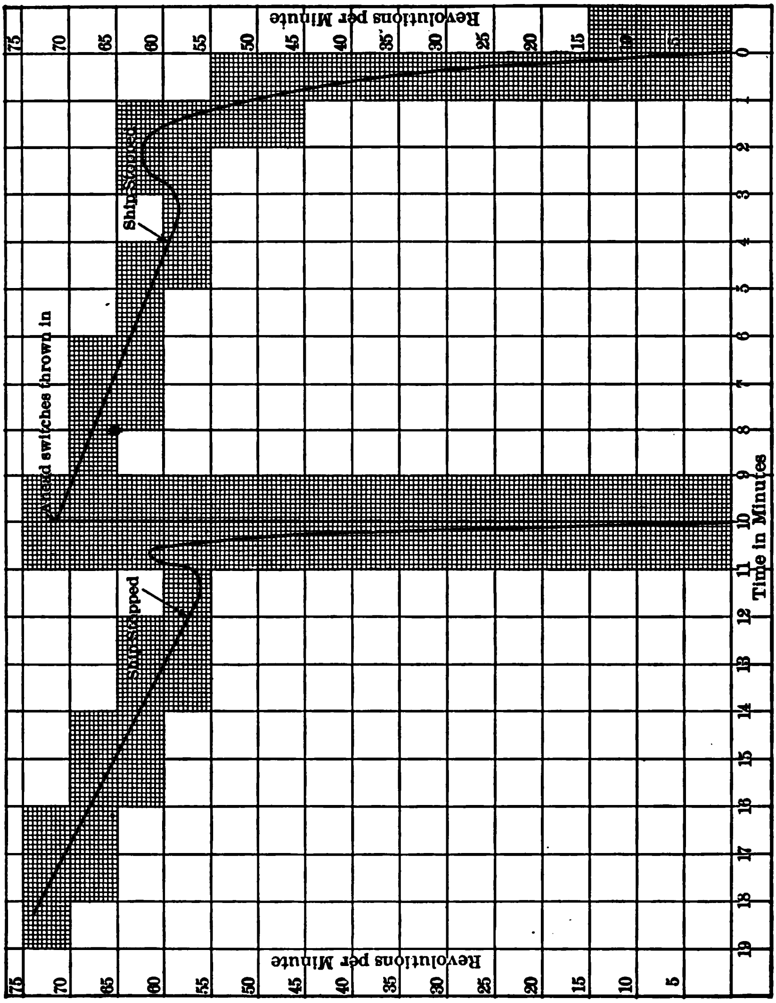

Revolutions per Minute
Ship Stopped
Ship Stopped
Time in Minutes
Revolutions per Minute

FIG. 58.

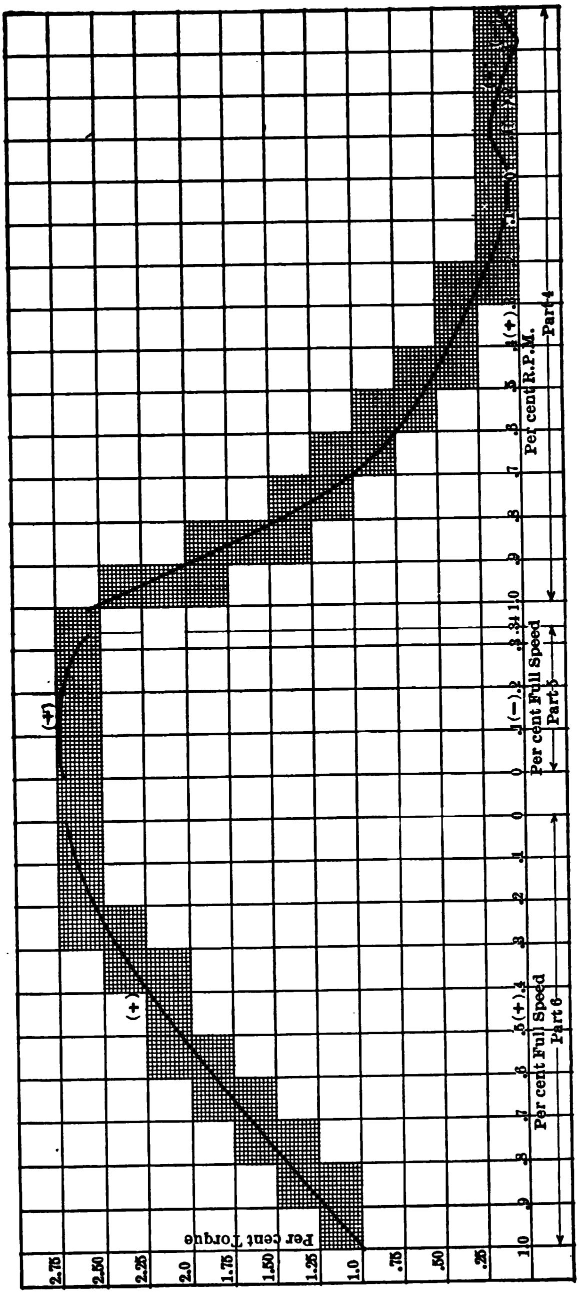


FIG. 57.—*Continued.*

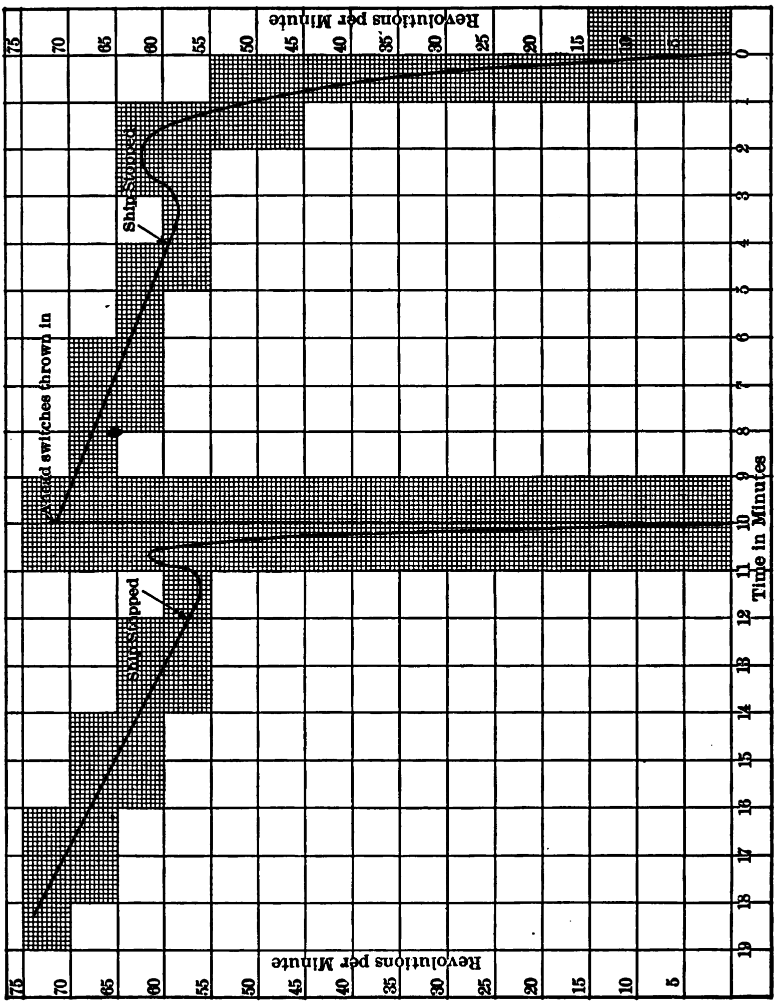

Revolutions per Minute
Time in Minutes
Ahead switches thrown in
Ship Stopped
Ship Stopped

FIG. 58.

CHAPTER XVI

MATERIALS FOR CONSTRUCTION OF AND GENERAL REQUIREMENTS FOR SCREW PROPELLERS

Material of Blades. The materials of which propeller blades are made are cast iron, cast steel, forged steel, manganese or some other strong bronze, and Monel metal.

Cast iron is used for the blades of propellers which work under conditions rendering them very liable to strike against obstructions. When so striking, the cast iron being weak, the blade breaks and, by so breaking, saves the shafting or the engine. Its disadvantages are extreme corrosion in sea water, heavy blade sections, and blunt edges due to the weakness of the metal.

Semi-steel is the name given to cast iron when a percentage of steel scrap has been added to the pig iron in the cupola. While stronger than cast iron, it is liable to carry flaws and is unreliable. Its use for propeller blades is not recommended.

Cast steel is stronger than cast iron, but has the same disadvantages although in a lesser degree.

Forged steel was formerly used in some instances for torpedo-boat propellers, but is not met with in present-day practice. It also possessed the disadvantage of excessive corrosion with consequent roughening and weakening of blade.

Manganese bronze and other strong bronzes appear to be all that may be wished for in propeller material. They are of high strength, permitting a low ratio of thickness to width of blade, can be brought to a sharp edge, and can be highly polished, while the corrosion to which they are subject is comparatively slight. They also cast without difficulty, giving blades free from porosity and blow holes. They may exercise a strong corrosive action on a steel hull if care is not taken to protect the hull in their vicinity by zinc plates.

Monel metal is extremely strong and tough, permits of very light blade sections and sharp edges, takes a very high polish, and is practically non-corrosive in sea water. These qualifications are all very desirable in a propeller metal. It has, however, the undesirable qualities of heavy and irregular warping of the blades when cooling in the mould, and a tendency to porosity around the tips and blade edges. On account of the tendency to warp, it is very difficult to insure the desired pitch unless the blade be cast with a large amount of waste metal which will permit of pitch correction in machining.

Material of the Hub. Hubs are usually made of cast iron or semi-steel for cast-iron or cast-steel propellers; of semi-steel for the poorer classes of work, and of manganese bronze for the better class, with manganese bronze blades; of manganese bronze or of Monel metal for Monel-metal blades. Where the propeller is cast solid, of course, the hub is of the same material as the blades.

General Requirements for Propellers

For all propellers except those made of cast iron or cast steel, the blades should be polished in order to reduce surface friction. With cast-steel and cast-iron blades, however, as they are usually used for work where the speeds of revolution and tip-speeds are low, the loss due to roughness of surface is not very high and it is preferable to retain the hard skin of the casting as a guard against corrosion than to sacrifice it in order to gain an advantage which would be only temporary.

Where blades and hub are made of one of the strong bronzes or of Monel metal, both blades and hub should be polished, the blades be made as thin as is consistent with strength, and the blade edges sharpened.

For work of the highest class and where the speeds of revolutions are high, the blades should be machined to true pitch, the backs of the blades finished to template and the blades polished to as smooth a surface as possible. The propeller should then be swung upon a mandrel and accurately balanced, as lack of

balance will produce excessive vibration when the speeds of revolution are high. In some cases, in order to insure a smooth blade surface, bronze blades have been silver-plated. This, however, seems of questionable expediency. It insures smooth blades for the trial trip, but it is doubtful if the silver-plating would remain on the blades for any considerable length of time.

CHAPTER XVII

GEOMETRY AND DRAUGHTING OF THE SCREW PROPELLER

GEOMETERY OF THE SCREW PROPELLER

THE geometrical construction of the screw propeller forms one of the most interesting problems for the engineer and the draughtsman; it is also of equal interest to the patternmaker and the foundryman, who are called upon to produce the structure itself from the plans. Therefore, a thorough understanding of the methods employed to generate its construction should be useful to all concerned.

The screw propeller of uniform pitch is the one which is generally accepted by engineers for the propulsion of ships, and this is divided into two distinct types, viz.: propellers having the vertical generatrix, and propellers having the inclined generatrix, the vertical generatrix being preferable for twin or other multiple screws where ample clearance between blade tips and hull, and between leading edges of the blades and the after edges of struts exists, while the inclined generatrix has advantages on single-screw vessels where the propeller is working behind the usual stern post.

In order to make the construction of the various types referred to as clear as possible, diagrams have been prepared which show the methods involved. Also, drawings of various propeller giveels have been made which conform to those diagrams, thus whing a very definite idea of the whole subject.

One of the simplest methods of making a screw propeller, viz.: that of sweeping up in the foundry, will probably afford a good illustration as to how a true screw may be generated. Suppose, as an example, a propeller of 12 ft. diameter and 12 ft. pitch be taken, which means that the propeller must make one complete

revolution in order to advance 12 ft. when no slip occurs. If a piece of paper be cut so that the base represents the circumference of the propeller, and the perpendicular represents its pitch, and this paper be wrapped around a cylinder whose diameter is 12 ft., the hypothenuse will generate the true helical line, Fig. 59.

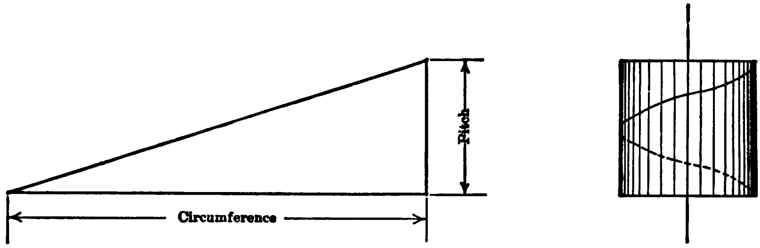


Fig. 59.

In a screw propeller, a fractional part only of the helix is dealt with, and this is used as the upper or guiding edge of an angle board from which to generate the working surface of one blade.

Instructions for Sweeping up the Helical Surface of a Screw Propeller

Make a level surface and lay off the center-lines, and also the outer radius of the wheel. Erect a cylindrical column on the intersection of the center-lines of this surface in a vertical position, then erect the angle board on this same surface at its proper radial distance out from the center of the column and parallel with the column. Construct a straight-edge having one end arranged to fit around and slide up and down this column, of sufficient length to extend beyond the angle board and provide means for keeping this straight-edge at an angle of 90° with the column at all times.

The surface of a screw propeller blade having a vertical generatrix can now be developed by simply causing the straight-edge to follow the helical edge of the angle board while passing through the arc *AB*, Fig. 60 and Fig. 61.

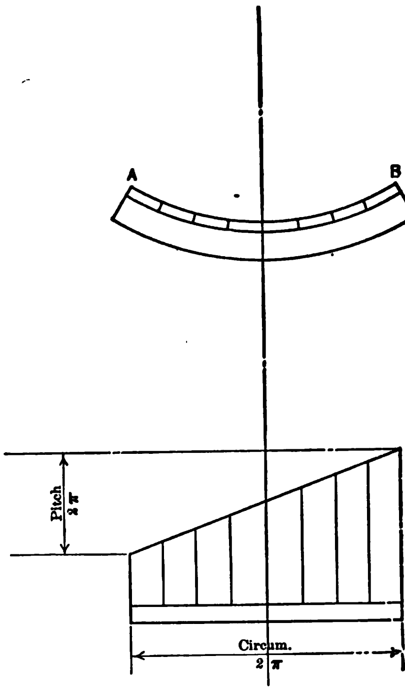


Fig. 60.

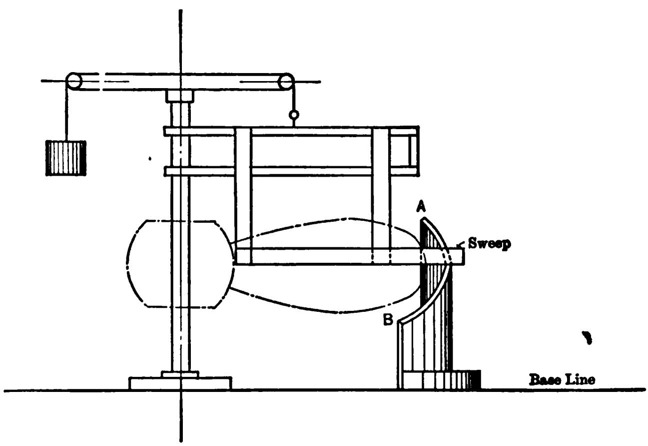


Fig. 61.

If the straight-edge be inclined at any angle other than 90° and the same operation be carried out, a screw having an inclined generatrix will be developed.

Geometry of a Screw Propeller having a Vertical Generatrix. Sheet 28

Lay down a base-line and erect a perpendicular to it. Where they intersect will be the axis on which the blade is generated.

On each side of this perpendicular draw lines, at an angle to suit the form and area of the projected surface, passing through the axis.

An angle of any magnitude may be used, but in this case 30° has been chosen, as this angle will cover any ordinary condition.

Through these 30° angles, draw an arc with a radius equal to the radius of the propeller. Divide this radius into ten equal parts, and draw in arcs through each of these division points. (See Fig. 1.) Divide the angles into five equal parts and draw lines passing through the axis. (Lines *a*, *b*, *c*, *d*, *e*, *f*, Fig. 1.) Where the arcs intersect the 30° angles draw lines parallel to the base-line. (Lines 1 to 10, Fig. 1.)

Now, to the right or left of this figure, erect another perpendicular to the base. This perpendicular becomes the generatrix on which the blade is developed.

The helix that the tip of the screw develops has already been explained. As 30°, or $\frac{1}{12}$ of the whole circumference, has been used to develop the blade, so $\frac{1}{12}$ of the pitch must be used. (If any angle other than 30° is used, the same proportion of the pitch must be used, and as $\frac{1}{12}$ of the circumference has been divided into five equal parts, so the $\frac{1}{12}$ pitch must also be divided.)

On each side of the generatrix lay off $\frac{1}{12}$ of the pitch, and divide this into 5 equal parts, and through these points erect perpendiculars passing through the base-line. (Lines *a*, *b*, *c*, *d*, *e*, *f*, Fig. 2.)

Fig. 4 is the plan view. The center-line in this view represents the plane through which the generatrix would move if rotated around its axis without any pitch. On each side of this

center-line project the $\frac{1}{12}$ of the pitch divided into 5 parts, as in Fig. 2. Draw lines parallel to the center-line (Lines *a, b, c, d, e, f*), which become the radial lines. Project up from Fig. 1, through the intersection of the line, *a, b, c, d, e, f*, with the arc of the full radius, to the lines *a, b, c, d, e, f*, respectively (Fig. 4), and draw in the curve passing through these intersections. This forms the helix.

The curves 1, 2, 3, 4, 5, 6, 7, 8, 9, Fig. 4, are formed by projecting up from the intersections of lines 1, 2, 3, 4, etc., Fig. 1, with the radial lines *a, b, c*, etc., Fig. 1, to the corresponding lines, *a, b, c, d*, etc., Fig. 4. The curves thus formed are true curves of the driving face at sections passed through lines 1, 2, 3, 4, etc., Fig. 1.

Geometry of a Screw Propeller having an Inclined Generatrix. Sheet 29

The surface having an inclined generatrix is developed similarly to the one having a vertical generatrix, except that, looking at the plan the sections will not pass through the same center on account of the inclination.

Draw the construction lines in Fig. 1, as for the vertical generatrix. Draw the generatrix at some predetermined angle (Fig. 2). Lay off the $\frac{1}{12}$ pitch, divided into 5 parts, each side of the generatrix at the tip and at the base. (Fig. 2, lines *a, b, c, d e, f*.) Draw perpendiculars to the tip-line passing through these points. Project from Fig. 1 the intersections of the arc of the full radius with lines *a, b, c, d, e, f*, to the corresponding lines, *a, b, c, d, e, f*, Fig. 2. Through these intersections draw in the curve. Now, from these intersections draw lines passing through points *a, b, c, d, e, f*, on the base-line. Then project the arcs of the tenths of the radius (Fig. 1), from where they intersect lines *a, b, c, d, e, f*, to the corresponding lines *a, b, c, d, e, f*, Fig. 2.

Lay down two lines parallel with each other at the distance "*A*" apart, and project around the points *a, b, c, d, e, f*, from Fig. 2, on both the tip and base-lines (see Fig. 4).

The radial lines *a*, *b*, *c*, *d*, *e*, *f*, in the plan must correspond to radial lines *a*, *b*, *c*, *d*, *e*, *f*, in Figs. 1 and 2. Project from the intersection of the arc of the full radius with lines *a*, *b*, *c*, *d*, *e*, *f*, Fig. 1, to the corresponding lines *a*, *b*, *c*, *d*, *e*, *f*, Fig. 4 laid off from the tip-line. A curve drawn through these points will form the helix. Through these points on the helix draw lines passing through the points *a*, *b*, *c*, *d*, *e*, *f*, laid off from the base-line along the center-line. The point on the center-line through which each of the sections, 1, 2, 3, 4, 5, 6, 7, 8, 9, 10, will pass will be the distance apart caused by the inclination.

These distances are obtained by projecting lines, 1, 2, 3, 4, etc., Fig. 1, through the generatrix, Fig. 2. Then the sine of the angle, formed by the generatrix and a perpendicular to the base passing through this intersection will be the distance to lay off from the base-line along the center-line in Fig. 4, and the point through which each of the sections, 1, 2, 3, 4, etc., will pass. The sections are then formed by passing curves through the intersections of lines *a*, *b*, *c*, *d*, *e*, *f*, Fig. 4, with the projections from the corresponding lines *a*, *b*, *c*, *d*, *e*, *f*, Fig. 1, where they intersect the lines 1, 2, 3, 4, etc., Fig. 1.

The Draughting of the Propeller

Sheet 30 shows a propeller of the type used on merchant vessels with a single screw working behind the stern post, and, in order to give a good clearance, it is made with an inclined generatrix which throws the tips of the blades further away from the post.

Sheet 31 shows a type of propeller used on torpedo boat destroyers, driven by turbines directly, where a high number of revolutions is necessary. The propeller shown is built with a vertical generatrix.

Sheet 32 shows a type of propeller used on battleships where the revolutions are comparatively low.

The following are the calculations and points of design, which are practically the same for any type of wheel, the principal dimensions for design having been calculated from Sheet 20.

Number of blades.................. 3
Diameter......................... 17′ 6″
Pitch........................... 18′ 0″
R.P.M.......................... 117
I.H.P_d...........................14,700—total 2 engines
I.H.P_d........................... 7350—one shaft
$\frac{\text{P.A.}}{\text{D.A.}}=.32.$

The geometrical construction is laid down for the vertical generatrix, and on this plot the form of projected surface. Sheet 32 is the sheet to which the following work applies:

From Sheet 25—Determine the chords of half arcs for $\frac{\text{P.A.}}{\text{D.A.}}$ $=.32-3$-bladed wheel. As no multipliers are given for ratio. 32 in the list interpolate by direct proportion, as follows:

Take the multiplier given for ratio .30 and increase it by the proportion $\frac{.32}{.30}$ for each chord of the several half arcs; this gives the following:

$\frac{.32}{.30}\times$ multiplier $\times$ rad. in inches $=\frac{.32}{.30}\times$multiplier$\times 105''$ $=112\times$multiplier; then

$.2\ \ R=112\times.082=9.184$ in.
$.3\ \ R=112\times.128=14.336$ in.
$.4\ \ R=112\times.170=19.04$ in.
$.5\ \ R=112\times.207=23.184$ in.
$.6\ \ R=112\times.236=26.432$ in.
$.7\ \ R=112\times.253=28.336$ in.
$.8\ \ R=112\times.250=28.00$ in.
$.9\ \ R=112\times.210=23.52$ in.
$.925R=112\times.190=21.28$ in.
$.95\ \ R=112\times.161=18.032$ in.
$.975R=112\times.120=13.44$ in.

Lay down these chords on each side of the center-line on the half arcs of the tenths of the radius and draw in the curve passing

through these points. Now, project the intersections of this curve with the lines 1, 2, 3, 4, etc., in the front elevation, Fig. 1, to the corresponding lines 1, 2, 3, 4, in the plan, Fig. 3, and side elevation, Fig. 2, and draw in the curve in these other two views, Figs. 2 and 3.

Determine the thickness of the blade at the root by the method described in chapter on Blade Thickness, as follows:

Blade, manganese bronze, 60,000 T.S.

Design based on Sheet 20.

T = Thickness of blade at root.

W = Width of blade tangent to hub $= 3'\ 4'' = 40''$.

$$A = \frac{33{,}000 \times \text{I.H.P.}}{2\pi \times R \times N} = \frac{33{,}000 \times 7350}{6.2832 \times 117 \times 3} = 110{,}000 \text{ ft.-lb.}$$

$B = .31 \times$ diam. in feet $=$ mean arm $= .31 \times 17.5 = 5.425$ ft.

$C = \dfrac{A}{B} = \dfrac{110{,}000}{5.425} = 20{,}270 =$ resultant athwartship force on one blade in foot-pounds.

$D = 12 \times B -$ Rad. of hub in inches $=$ arm of athwartship force measured to root of blade $= 12 \times 5.425 = 65.1'' - 25{,}5'' = 39.6''$.

$E = C \times D =$ athwartship moment at root of blade in inch-pounds $= 20{,}270 \times 39.6 = 802{,}800$.

$F = \dfrac{33{,}000 \times \text{I.H.P.}}{\text{Pitch} \times R \times \text{No. blades}} =$ indicated thrust per blade in pounds

$$= \frac{33{,}000 \times 7350}{18 \times 117 \times 3} = 38{,}390 \text{ lb.}$$

$G = .345 \times$ diam. in inches $=$ mean arm of thrust $= 345 \times 210 = 72.45$ in.

$H = G -$ Rad. hub in inches $=$ arm of thrust measured to the root of blade $= 72.45 - 25.5 = 46.95$.

$J = F \times H =$ fore and aft moment at root of blade, in inch-pounds $= 38{,}390 \times 46.95 = 1{,}802{,}400$.

$K = \dfrac{\text{Circ. hub ft.}}{\text{Pitch ft.}} =$ tangent of angle between face of blade and C.L. of hub or fore-and-aft line tangent to surface of hub

$$= \frac{13.35}{18} = .7418.$$

L = Sine of arc whose tangent is $K = 36° - 34' =$ Sine .5958.
M = Cosine of arc whose tangent is $K = 36° - 34' =$ cosine .8032.
$N = L \times J$ = component of fore and aft moment normal to face of blade at root $= .5958 \times 1{,}802{,}400 = 1{,}009{,}100$.
$O = M \times E$ = same for athwartship moment $= .8032 \times 802{,}800 = 644{,}800$.
$P = N + O$ = total moment at root of blade in inch pounds $= 1{,}009{,}100 + 644{,}800 = 1{,}653{,}900$.
f = Fibre stress = 10,000 as e.h.p. ÷ E.H.P. = 1.0.

$$T = \sqrt{\frac{P \times 13{,}125}{W \times f}} = \sqrt{\frac{1{,}653{,}900 \times 13{,}125}{40 \times 10{,}000}} = 7.367'';$$

say $7\frac{1}{2}''$ for safety.

Draw in the shape of the back of the blade, as in the section shown on Fig. 2, Sheet 32, using 7.5 in. at the root and a thickness of $\frac{3}{4}$ in. at the tip. The tip is then fined down to a very small radius, about $\frac{1}{4}$ in. at the tip and faired back to about 5 in. from the edge of the blade. Now, determine and draw in the flange of the blade. This is determined from the plan view, Fig. 3, and must be of a diameter large enough to take the blade and the blade bolts. The diameter of the hub can now be determined by drawing in the flange in the front elevation, Fig. 1, and drawing a circle, with the axis of the blade as the center and the radius of the hub forming the top of the flange.

The hubs of built-up propellers are usually spherical, as in this case.

The number and size of the bolts holding the flange to the hub must now be determined.

The number used is dependent upon the space on the flange to accommodate them, but either 7 or 9 is the most common practice. Nine have been used in this case, and are spaced 5 on the driving side of the blade and 4 on the backing side

The area of the bolts is determined in the following manner:

A = area in square inches of one bolt.
n = Number of bolts on driving side of one blade = 5.
r = Rad. of pitch circle (as the leverage is different for each bolt, the radius is taken as the mean) = $14''$.

L = arm of thrust measured from face of flange = .345 × diam. − distance from C.L. of hub to face of flange = (.345 × 210) − $13\frac{1}{2}$ = 58.95 in.

N = Number of blades = 3.

R = Revs. per minute = 117.

P = Pitch in feet = 18.

f = Stress—manganese bronze or naval brass = 6,000 lb.

$$A = \frac{\text{I.H.P.} \times 33{,}000 \times L}{N \times P \times R \times n \times r \times f} = \frac{7350 \times 33{,}000 \times 58.95}{3 \times 18 \times 117 \times 5 \times 14} \times 6000 =$$

5.388 sq. in. = $2\frac{5}{8}$ in. diameter.

(As 6000 lb. is max. stress to be used,—bolts have been made 3 in. diameter = area 7.0686, and the stress is:

$$\frac{7350 \times 33{,}000 \times 58.95}{3 \times 18 \times 117 \times 5 \times 14 \times 7.0686} = 4574 \text{ lb.}$$

Care must be taken in spacing the bolts in the flange that the section of the blade where it joins the hub, fillets being neglected, is not decreased in area. This can easily be avoided by projecting, in the plan view, Fig. 3, the line of intersection of the driving face and the back of the blade, with the surface of the hub, as shown by the dashed lines.

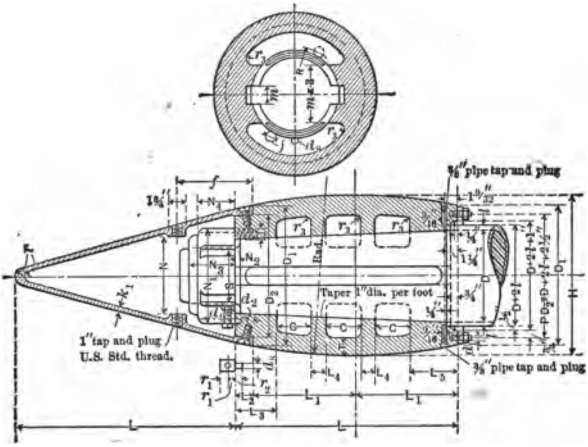
The blade shown on Sheet 32 has the bolt holes slotted to allow for an adjustment of the pitch 9 in. either way, but this practice is, while allowable, incorrect, as the propeller is no longer a true screw if the pitch be altered from the designed pitch.

The practice in the U. S. Navy is to provide a cover over the bolts and nuts, flush with the surface of the flange, and having a water-tight joint with the flange.

The bolts are locked in place by locking pieces, which fit between the heads and are held in place by a tap bolt screwed into the flange between the bolts.

STANDARD PROPELLER HUBS

For Solid Propellers—Plate B



D	D_1	D_2	H	L	L_1	L_2	L_3	L_4	L_5	N	N_1	N_2	N_3	N_4	PD	R	S
Above 3– 4" incl.	5¾	4¾	6⅝	9¼	4⅛	1	....	....	....	3¼	4⅛	...	1 15/16	1½	Made to suit D and l	2	2¼
4– 5	7⅝	6¼	8½	12	5⅜	1¼	....	....	....	4⅜	5¼	...	2 3/16	2 3/16		2 3/16	3
5– 6	9⅜	7¾	10½	14½	6½	1½	....	....	....	5⅛	6⅛	...	3	2⅜		3 3/16	3½
6– 8	11¾	10¼	13½	18½	8⅜	1¾	3 7/16	1⅛	4 3/16	6½	7¾	½	3⅞	3 1/16		4	4½
8–10	15¼	13¼	17½	24	11 1/16	1⅞	4⅜	1½	5⅛	8¾	10¼	½	5⅛	4⅛		5 3/16	6
10–12	18¾	16¾	21½	29	13½	2	5 5/16	1 13/16	5 15/16	10⅝	12¼	⅝	6¼	5		6⅜	7¼
12–14	22¼	20	25½	34½	16¾	2⅛	6⅜	2 3/16	6⅞	12⅜	14¼	¾	7¼	5⅞		7½	8½
14–16	25¼	22½	29	39¾	18¾	2¼	7⅜	2½	7⅝	14⅝	16¾	1	8½	6 13/16		8⅞	10
16–18	28¾	26	33	45	21 5/16	2⅜	8 5/16	2 15/16	8 7/16	16⅜	18¾	1⅛	9½	7 13/16		10	11¼
18–20	32¼	29¼	37	50½	24	2½	9 3/16	3 3/16	9 5/16	18¼	20¾	1⅛	10⅝	8⅝		11 1/16	12½
20–22	35¼	32¼	40½	55¾	26⅝	2½	10 3/16	3½	10 3/16	20½	23¼	1¼	11⅞	9 11/16		12 3/16	14
22–24	38½	35¼	44½	61	29 3/16	2⅝	11⅛	3 13/16	11⅛	22¼	25¾	1¼	13	10½		13¼	15¼

D	T	d_1	d_2	No. of d_2	d_8	d_9	f	j	k_1	l	m	r	r_1	r_2	r_3
Above 3– 4" incl.	...	⅝	⅜	6	5/16	⅜	3¼	½	⅛	3" – 5 " inc. l = ¼"	From Bureau Design Formulas see Sec. II and VI, Sub. 5. sh. 45.	5/16	½	⅜	..
4– 5	...	⅝	½	6	⅜	7/16	4⅞	⅝	5/32	5" – 7½" inc. l = ⅜"		⅜	⅝	7/16	..
5– 6	...	⅝	⅝	6	⅜	½	5⅞	¾	3/16	7½"–10 " inc. l = ½"		½	⅝	7/16	..
6– 8	1 5/16	⅝	⅝	6	7/16	½	7	1	7/32	10 "–12½" inc. l = ⅝"		⅝	¾	½	⅝
8–10	2	⅝	¾	8	9/16	⅝	8¼	1¼	¼	12½"–15 " inc. l = ¾"		13/16	⅞	⅝	¾
10–12	2½	⅝	¾	8	⅝	⅝	9½	1½	5/16	15 "–17½" inc. l = ⅞"		1	1	¾	⅞
12–14	3	⅝	⅞	10	¾	¾	10¾	1¾	⅜	17½"–20 " inc. l = 1"		1⅛	1 3/16	⅞	1
14–16	3⅜	¾	⅞	10	13/16	1	12	2	7/16	20 "–24 " inc. l = 1⅛"		1⅜	1 5/16	15/16	1⅛
16–18	3¾	¾	1	10	15/16	1	13¼	2¼	15/32	Shaft dia. D above		1½	1½	1 1/16	1¼
18 20	4¼	¾	1	10	1	1	14½	2½	½			1⅝	1⅝	1⅛	1⅜
20–22	4⅝	¾	1	12	1⅛	1⅛	15¾	3	9/16			1⅞	1 13/16	1 3/16	1½
22–24	5⅛	¾	1⅛	12	1 3/16	1⅛	17	3¼	⅝			2	1⅞	1⅜	1⅝

STANDARD PROPELLER HUBS

FOR 4-BLADE, BUILT-UP, PROPELLERS—PLATE C

FORMULAS FOR OBTAINING *d*.

***A = net area, sq. in., one bolt. a = distance from center of shaft to center of thrust, in inches. b = distance from center of shaft to face of blade flange. D = diameter of propeller in inches. f = allowed stress in lbs. per sq. in. of bolt section. G = arm of thrust measured from face of flange, in inches. N = number of blades. n = number of bolts on driving side of one blade. P = pitch in feet. R = revolutions per minute. r = radius of pitch circle in inches.**

$$A = \frac{\text{S.H.P.} \times 33000 \times G}{N \times P \times R \times n \times r \times f}$$

$$a = D \times .345,$$

$$G = a - b,$$

f = 4500 for manganese bronze or naval brass.

D	B	C	D_1	D_2	D_3	F	F_1[0]	F_2	H	K	L	L_1	L_2	N
8¼	7 [illegible]	6	12¼	15 11/16	13⅛	14⅛	28	5⅞	24	9 1/16	20½	10 7/16	10 1/16	8
Above 8¼–9 incl. .	8¼	6½	13¼	17⅛	14¼	15 5/16	28	6⅜	26	9 11/16	21¾	11¼	10½	8¾
9–9½	8⅝	7	14¼	18 7/16	15 5/16	16½	28	6⅞	28	10 5/16	23	12⅛	10⅞	9⅜
9½–10¼	9½	7½	15¼	19¾	16⅜	17 11/16	28	7⅜	30	11 1/16	24½	13	11½	9⅞
10¼–11	10⅛	8	16 3/16	21 1/16	17½	18⅞	28	7⅞	32	12 1/16	26⅛	13⅞	11¾	10¼
11–11½	10¾	8½	17 5/16	22⅜	18 9/16	20	28	8⅜	34	12 11/16	27¾	14¾	13	11
11½–12¼	11⅝	9	18 5/16	23 11/16	19⅝	21 3/16	28	8⅞	36	13 5/16	29 7/16	15⅝	13 13/16	11⅝
12¼–13	12	9½	19⅜	25	20¾	22⅜	28	9 5/16	38	14 1/16	31	16 7/16	14 9/16	12⅜
13–13¾	12⅝	10	20⅜	26 5/16	21⅞	23 9/16	28	9 13/16	40	15⅛	32 11/16	17⅜	15 5/16	13⅛
13¾–14¼	13¼	10½	21⅜	27⅝	22 15/16	24¾	28	10 5/16	42	15 13/16	34 5/16	18 3/16	16⅛	13⅞
14¼–15	13 13/16	11	22 7/16	28 15/16	24 1/16	25 15/16	28	10 13/16	44	16⅝	35 15/16	19 1/16	16⅞	14¼
15–15¾	14 9/16	11½	23 7/16	30¼	25⅛	27⅛	28	11 5/16	46	17⅜	37 9/16	19 15/16	17⅝	15
15¾–16½	15 5/16	12	24 7/16	31 9/16	26¼	28¼	28	11 13/16	48	18⅛	39¼	20⅞	18⅜	15¾
16½–17	15 15/16	12½	25½	32⅞	27 5/16	29 7/16	28	12¼	50	18⅞	40 13/16	21 11/16	19⅛	16⅜
17–17¾	16 7/16	13	26½	34¼	28 7/16	30⅝	28	12 13/16	52	19⅝	42½	22 9/16	19 15/16	16¾
17¾–18½	17 1/16	13½	27½	35 9/16	29 9/16	31 13/16	28	13¼	54	20⅜	44⅛	23 7/16	20 11/16	17½
18½–19¼	17 11/16	14	28½	36⅞	30⅝	33	28	13¾	56	21⅛	45¾	24 5/16	21 7/16	18¼
19¼–20	18⅜	14½	29 9/16	38 3/16	31 11/16	34 3/16	28	14¼	58	21⅞	47⅜	25⅛	22¼	19
20–20½	19	15	30 9/16	39½	32 13/16	35⅜	28	14¾	60	22⅝	49	26	23	19¾
20½–21¼	19⅝	15½	31⅝	40 13/16	33⅞	36½	28	15¼	62	23⅜	50⅝	26⅞	23¾	20⅜
21¼–22	20¼	16	32⅝	42⅛	35	37 11/16	28	15¾	64	24⅛	52¼	27¾	24½	20⅞
22–22½	20⅞	16½	33⅝	43 7/16	36 1/16	38⅞	28	16¼	66	24⅞	53 13/16	28⅝	25 5/16	21½

D	N_1	N_2	N_3	N_4	PD	PD_1	PD_2	R	S	T	T_1	T_2	T_3	W
8¼	9⅜	½	4⅝	3 13/16	11¾	13⅜	Made to suit diameter D and thickness L.	4¾	5½	1½	1 15/16	¼	2 11/16	2
Above 8¼–9 incl. .	9¾	½	5	4	12¾	14½		5⅛	5¾	1⅝	2⅛	¼	2 15/16	2 3/16
9–9½	10⅝	½	5⅜	4 5/16	13 11/16	15⅝		5 9/16	6¼	1¾	2¼	¼	3⅛	2⅜
9½–10¼	11⅛	½	5¾	4 11/16	14 11/16	16¾		5 15/16	6¾	1⅞	2 7/16	5/16	3⅜	2½
10¼–11	11⅞	½	6⅛	4⅞	15 11/16	17⅞		6 5/16	7	2	2⅝	5/16	3 9/16	2 11/16
11–11½	12⅝	⅝	6⅜	5 3/16	16⅝	19		6¾	7½	2⅛	2¾	5/16	3 13/16	2⅞
11½–12¼	13½	¾	6¾	5 9/16	17⅝	20⅛		7⅛	8	2¼	2 15/16	⅜	4 1/16	3
12¼–13	14¼	¾	7¼	5⅞	18⅝	21 3/16		7½	8½	2⅜	3⅛	⅜	4¼	3 3/16
13–13¾	15⅛	¾	7⅝	6¼	19 9/16	22 5/16		7 15/16	9	2½	3¼	⅜	4½	3⅜
13¾–14¼	15½	¾	7⅞	6½	20 9/16	23 7/16		8 5/16	9¼	2⅝	3 7/16	7/16	4¾	3½
14¼–15	16⅜	⅞	8¼	6¾	21 9/16	24 9/16		8 11/16	9¾	2¾	3 9/16	7/16	4 15/16	3 11/16
15–15¾	17⅛	1	8 11/16	7 1/16	22½	25 11/16		9⅛	10¼	2⅞	3¾	7/16	5 3/16	3⅞
15¾–16½	18	1	9⅛	7 7/16	23½	26 13/16		9½	10¾	3	3 15/16	½	5⅜	4
16½–17	18¾	1⅛	9½	7 11/16	24½	27 15/16		9⅞	11¼	3⅛	4 1/16	½	5⅝	4 3/16
17–17¾	19¼	1⅛	9¾	7 13/16	25½	29		10 5/16	11½	3¼	4¼	½	5 13/16	4⅜
17¾–18½	20	1⅛	10 3/16	8 5/16	26 7/16	30⅛		10 11/16	12	3⅜	4⅜	9/16	6 1/16	4½
18½–19¼	20¾	1⅛	10⅝	8⅝	27 7/16	31¼		11 1/16	12½	3½	4 9/16	9/16	6 5/16	4 11/16
19¼–20	21⅝	1¼	11⅛	9	28⅜	32⅜		11½	13	3⅝	4¾	9/16	6½	4⅞
20–20½	22½	1¼	11½	9⅜	29⅜	33½		11⅞	13½	3¾	4⅞	⅝	6¾	5
20½–21¼	22⅞	1¼	11 11/16	9½	30⅜	34⅝		12¼	13¾	3⅞	5 1/16	⅝	6 15/16	5 3/16
21¼–22	23⅝	1¼	12¼	9⅞	31⅜	35¾		12⅝	14¼	4	5¼	⅝	7 3/16	5⅜
22–22½	24½	1¼	12½	10 5/16	32 5/16	36⅞		13 1/16	14¾	4⅛	5⅜	11/16	7⅜	5½

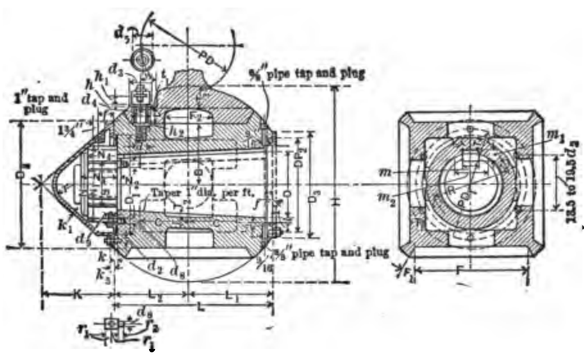


PLATE C

STANDARD PROPELLER HUBS—*Continued*

D	d	d_1	d_2	d_3	d_4	d_5	d_8	d_9	f	f_1	h	h_1	h_2
8½	* From bureau design formulas Sec. VII Subj. 57 Sh. 5	⅝	⅝	1.9d	1.2d	1.5d+⅛	½	½	1½	1⅞	.233d	.766d	.166d
Above 8½–9 incl.		⅝	⅝	1.9d	1.2d	1.5d+⅛	9/16	9/16	1½	2 1/16	.233d	.766d	.166d
9–9⅝		⅝	⅝	1.9d	1.2d	1.5d+⅛	9/16	⅝	1½	2¼	.233d	.766d	.166d
9⅝–10¼		⅝	⅝	1.9d	1.2d	1.5d+⅛	⅝	⅝	1½	2⅜	.233d	.766d	.166d
10¼–11		⅝	⅝	1.9d	1.2d	1.5d+⅛	⅝	⅝	1½	2½	.233d	.766d	.166d
11–11⅝		⅝	⅝	1.9d	1.2d	1.5d+⅛	⅝	⅝	1½	2 11/16	.233d	.766d	.166d
11⅝–12¼		⅝	⅝	1.9d	1.2d	1.5d+⅛	11/16	¾	1½	2⅞	.233d	.766d	.166d
12¼–13		⅝	⅝	1.9d	1.2d	1.5d+⅛	¾	¾	1½	3	.233d	.766d	.166d
13–13¾		⅝	¾	1.9d	1.2d	1.5d+⅛	¾	⅞	1½	3 3/16	.233d	.766d	.166d
13¾–14¼		⅝	¾	1.9d	1.2d	1.5d+⅛	¾	⅞	1½	3⅜	.233d	.766d	.166d
14¼–15		⅝	¾	1.9d	1.2d	1.5d+⅛	13/16	⅞	1½	3½	.233d	.766d	.166d
15—15¾		¾	¾	1.9d	1.2d	1.5d+⅛	⅞	1	1½	3⅝	.233d	.766d	.166d
15¾–16⅛		¾	¾	1.9d	1.2d	1.5d+⅛	⅞	1	1½	3 13/16	.233d	.766d	.166d
16⅛–17		¾	⅞	1.9d	1.2d	1.5d+⅛	15/16	1	1½	4	.233d	.766d	.166d
17–17¾		¾	⅞	1.9d	1.2d	1.5d+⅛	15/16	1	1½	4⅛	.233d	.766d	.166d
17¾–18⅛		¾	⅞	1.9d	1.2d	1.5d+⅛	1	1	1½	4¼	.233d	.766d	.166d
18⅛–19¼		¾	⅞	1.9d	1.2d	1.5d+⅛	1	1	1½	4 7/16	.233d	.766d	.166d
19¼–20		¾	⅞	1.9d	1.2d	1.5d+⅛	1 1/16	1⅛	1½	4 9/16	.233d	.766d	.166d
20–20⅝		¾	⅞	1.9d	1.2d	1.5d+⅛	1 1/16	1⅛	1½	4¾	.233d	.766d	.166d
20⅝–21¼		¾	⅞	1.9d	1.2d	1.5d+⅛	1 1/16	1⅛	1½	4⅞	.233d	.766d	.166d
21¼–22		¾	1	1.9d	1.2d	1.5d+⅛	1⅛	1⅛	1½	5 1/16	.233d	.766d	.166d
22–22⅝		¾	1	1.9d	1.2d	1.5d+⅛	1⅛	1⅛	1½	5¼	.233d	.766d	.166d

D	j	k	k_1	k_5	l	m	m_1	r	r_1	r_2	t	t_1
8½	1⅜	½	7/32	7/16	Above shaft dia. { 15″–17½″ incl. l = ⅞″; 17½″–20″ incl. l = 1″; 20″–22½″ incl. l = 1¼″ }	From Bureau Design Formulas, Sec. II and VI, Sub. 5, Sh. 45	2m	2 11/16	11/16	9/16	1⅞	2 1/16
Above 8½–9 incl.	1½	½	¼	½			....	2 15/16	⅞	⅝	2 1/16	2¼
9–9⅝	1⅝	½	¼	9/16			....	3⅛	⅞	⅝	2¼	2⅜
9⅝–10¼	1⅝	½	9/32	9/16			....	3⅜	1	¾	2⅜	2 9/16
10¼–11	1⅝	⅝	5/16	⅝			....	3⅝	1	¾	2½	2¾
11–11⅝	1⅝	⅝	5/16	⅝			....	3 13/16	1	¾	2 11/16	2⅞
11⅝–12¼	1¾	¾	11/32	11/16			....	4 1/16	1⅛	13/16	2⅞	3 1/16
12¼–13	1⅞	¾	11/32	¾			....	4¼	1 3/16	⅞	3	3¼
13–13¾	1⅞	⅞	⅜	¾			....	4½	1 3/16	⅞	3 3/16	3 7/16
13¾–14¼	2	⅞	⅜	13/16			....	4¾	1 5/16	⅞	3⅜	3 9/16
14¼–15	2	1	13/32	13/16			....	5	1 5/16	15/16	3½	3¾
15–15¾	2¼	1	13/32	⅞			....	5 3/16	1⅜	1	3⅝	3 15/16
15¾–16⅛	2¼	1⅛	7/16	15/16			....	5 7/16	1⅜	1	3 13/16	4⅛
16⅛–17	2½	1⅛	15/32	15/16			....	5⅝	1½	1 1/16	4	4¼
17–17¾	2½	1¼	15/32	1			....	5⅞	1½	1 1/16	4⅛	4 7/16
17¾–18⅛	2½	1¼	½	1			....	6 1/16	1⅝	1⅛	4¼	4⅝
18⅛–19¼	2¾	1⅜	½	1 1/16	Above shaft dia. { 7½″–10″ incl. l = ½″; 10″–12½″ incl. l = ⅝″; 12½″–15″ incl. l = ¾″ }		....	6 5/16	1⅝	1⅛	4 7/16	4¾
19¼–20	2¾	1⅜	17/32	1⅛			....	6½	1 11/16	1¼	4 9/16	4 15/16
20–20⅝	2¾	1½	17/32	1⅛			....	6¾	1 11/16	1¼	4¾	5⅛
20⅝–21¼	3	1½	9/16	1 3/16			....	7	1 11/16	1¼	4⅞	5 5/16
21¼–22	3	1⅝	9/16	1¼			....	7¼	1 11/16	1 5/16	5 1/16	5 7/16
22–22⅝	3	1⅝	19/32	1¼			....	7 7/16	1 11/16	1 5/16	5¼	5⅝

STANDARD PROPELLER HUBS

For 3-Blade, Built-up, Propellers—Plate D

D	*B*	*C*	*D*₁	*D*₂	*F*	*F*₁°	*F*₂	*H*	*K*	*L*	*L*₁	*N*	*N*₁	*N*₂
8 1/2	6 7/8	4 11/16	12 1/8	15	16 7/16	28	7 13/16	26	10 5/16	20 1/2	10 1/4	8	9 5/8	1/2
Above 8 1/2–9 incl..	7 3/8	5 1/4	13 1/4	16 7/16	18 5/16	28	8 5/8	28	11 1/4	21 3/4	10 7/8	8 5/8	9 3/4	1/2
9– 9 1/2	8	5 5/8	14 1/4	17 5/8	19 7/8	28	9	30	12	23	11 1/2	9 1/4	10 3/8	1/2
9 1/2–10 1/4	8 5/16	6	15 1/8	18 3/4	20 3/4	28	9 5/16	32	12 7/8	24 1/2	12 1/4	9 7/8	11 1/2	5/8
10 1/4–10 3/4	9 1/8	6 3/8	16 1/8	20	21 1/2	28	10 1/8	34	13 5/8	26	13	10 1/2	11 7/8	5/8
10 3/4–11 1/2	9 5/8	6 3/4	17	21 1/8	23 3/16	28	10 3/4	36	14 1/2	27 1/4	13 7/8	11	12 5/8	5/8
11 1/2–12	10 3/16	7 1/8	18	22 1/4	24 5/8	28	11 5/8	38	15 1/4	29 1/2	14 5/8	11 5/8	13 1/2	3/4
12 –12 3/4	10 3/4	7 1/2	18 15/16	23 3/8	25 3/4	28	12	40	16	30 3/4	15 3/8	12	13 7/8	3/4
12 3/4–13 1/2	11 1/4	7 7/8	19 7/8	24 5/8	27 5/16	28	12 5/16	42	16 7/8	32 1/4	16 1/8	12 3/4	14 3/4	3/4
13 1/2–14	11 11/16	8 1/4	20 13/16	25 13/16	28 3/16	28	13 1/8	44	17 5/8	33 3/4	16 7/8	13 1/4	15 1/2	3/4
14 –14 3/4	12 3/8	8 5/8	21 3/4	26 15/16	29 11/16	28	13 3/4	46	18 1/2	35 1/2	17 3/4	14 1/4	16 3/8	7/8
14 3/4–15 1/4	12 7/8	9	22 3/4	28 1/16	31 3/16	28	14 3/8	48	19 1/4	37	18 1/2	14 5/8	16 3/4	1
15 1/4- 15 3/4	13 3/8	9 3/8	23 5/8	29 3/8	32 7/16	28	14 13/16	50	20	38 1/2	19 1/8	15 1/8	17 1/2	1
15 3/4–16 1/2	13 13/16	9 3/4	24 5/8	30 1/2	33 13/16	28	15 5/16	52	20 7/8	39 3/4	19 7/8	15 3/4	18	1 1/8
16 1/2–17 1/4	14 1/2	10 1/8	25 1/2	31 11/16	35 1/16	28	16 1/8	54	21 5/8	41 1/2	20 3/4	16 5/8	18 3/4	1 1/8
17 1/4–18	15	10 1/2	26 1/2	32 7/8	36 5/16	28	16 3/4	56	22 1/2	43 1/4	21 5/8	17 1/4	19 5/8	1 1/8
18 –18 1/2	15 5/16	10 7/8	27 7/16	34	37 9/16	28	17 3/16	58	23 1/4	44 3/4	22 3/8	17 7/8	20 3/8	1 1/8
18 1/2–19 1/4	16 3/16	11 1/4	28 3/8	35 3/16	38 7/8	28	17 13/16	60	24	46 1/2	23 1/8	18 5/8	21 1/4	1 1/8
19 1/4–19 3/4	16 5/8	11 5/8	29 3/8	36 5/16	40 3/16	28	18 3/8	62	24 7/8	47 3/4	23 7/8	19	21 5/8	1 1/8
19 3/4–20 1/2	17 1/8	12	30 1/4	37 7/16	41 7/16	28	19 1/8	64	25 5/8	49 1/4	24 5/8	19 3/4	22 1/2	1 1/8
20 1/2–21	17 11/16	12 3/8	31 1/4	38 11/16	42 3/4	28	19 11/16	66	26 1/2	50 3/4	25 3/8	20 1/2	23 1/4	1 1/8
21 –21 3/4	18 1/4	12 3/4	32 3/16	39 7/8	44 1/4	28	20 5/16	68	27 1/4	52 1/4	26 1/8	20 7/8	23 5/8	1 1/8
21 3/4–22 1/2	18 3/4	13 1/8	33 1/8	41 1/16	45 5/8	28	20 13/16	70	28	53 3/4	26 7/8	21 7/8	24 7/8	1 1/8

D	*N*₃	*N*₄	*PD*	*PD*₁	*PD*₂	*R*	*R*₁	*R*₂	*S*	*T*	*T*₁	*T*₂	*T*₃	*W*	*W*₁
8 1/2	4 5/8	3 13/16	13 3/4	13 3/8	Made to suit dia. *D* and thickness *l*	4 3/4	4 5/16	8 13/16	5 3/8	1 13/16	2 3/16	2 3/8	2 1/2	3 3/8	1 3/4
Above 8 1/2–9 incl..	5	4	15	14 3/4		5 3/16	5	9 3/4	5 3/4	2 1/8	2 3/8	2 5/8	2 11/16	3 11/16	1 7/8
9– 9 1/2	5 3/8	4 3/16	16	15 3/4		5 5/16	5 5/8	10 7/16	6 1/4	2 1/4	2 9/16	2 13/16	2 7/8	3 13/16	2
9 1/2–10 1/4	5 5/8	4 11/16	17 1/8	16 11/16		5 11/16	5 11/16	11 1/8	6 5/8	2 3/8	2 11/16	3	3 1/16	4 1/16	2 1/8
10 1/4–10 3/4	6 1/8	4 7/8	18 1/4	17 7/8		6 5/16	6 1/16	11 13/16	7	2 9/16	2 7/8	3 3/16	3 1/4	4 1/4	2 1/4
10 3/4–11 1/2	6 3/8	5 3/16	19 3/8	18 11/16		6 5/8	6 7/16	12 3/8	7 3/8	2 11/16	3 1/16	3 3/8	3 1/2	4 1/2	2 5/16
11 1/2- 12	6 3/4	5 5/16	20 5/8	20		7	6 3/4	13 1/4	8	2 13/16	3 1/4	3 9/16	3 5/8	5	2 9/16
12 –12 3/4	7	5 11/16	21 3/16	21 1/16		7 5/8	7 1/8	13 13/16	8 1/4	3	3 3/8	3 3/4	3 7/8	5 1/4	2 11/16
12 3/4–13 1/2	7 3/8	6 1/16	22 1/8	22 1/8		7 3/4	7 3/8	14 5/8	8 5/8	3 1/8	3 9/16	3 13/16	4 1/16	5 1/2	2 13/16
13 1/2–14	7 7/8	6 3/8	23 5/16	23 1/8		8 1/8	7 7/8	15 5/16	9 1/4	3 5/16	3 3/4	4 1/8	4 1/4	5 11/16	2 15/16
14 –14 3/4	8 1/4	6 3/4	24 5/8	24 3/16		8 3/8	8 3/16	16	9 5/8	3 7/16	3 7/8	4 5/16	4 7/16	6 1/16	3 1/16
14 3/4–15 1/4	8 3/8	6 15/16	25 13/16	25 1/4		8 7/8	8 9/16	16 5/8	10	3 9/16	4 1/16	4 1/2	4 5/8	6 5/16	3 1/4
15 1/4–15 3/4	8 3/4	7 1/4	26 3/8	26 5/16		9 1/4	8 15/16	17 7/16	10 3/8	3 5/8	4 1/4	4 11/16	4 13/16	6 9/16	3 3/8
15 3/4–16 1/2	9 1/8	7 7/16	27 7/8	27 3/8		9 5/8	9 1/4	18 1/8	10 3/4	3 7/8	4 7/16	4 7/8	5	6 13/16	3 1/2
16 1/2–17 1/4	9 3/8	7 13/16	28 13/16	28 7/16		10	9 5/8	18 13/16	11 1/4	4	4 9/16	5 1/16	5 3/16	7 1/8	3 5/8
17 1/4–18	10	8 1/8	30	29 3/8		10 3/8	10	19 3/8	11 3/4	4 3/16	4 3/4	5 1/4	5 3/8	7 3/8	3 3/4
18 –18 1/2	10 3/8	8 3/8	31 3/16	30 3/8		10 3/4	10 3/8	20 3/16	12 1/4	4 5/16	4 15/16	5 7/16	5 9/16	7 5/8	3 7/8
18 1/2–19 1/4	10 5/8	8 11/16	32 1/4	31 5/16		11 1/16	10 11/16	20 7/8	12 3/4	4 3/8	5 1/16	5 5/8	5 13/16	7 7/8	4
19 1/4–19 3/4	11 1/8	9	33 1/4	32 5/8		11 7/16	11 1/16	21 9/16	13	4 5/8	5 1/4	5 13/16	6	8 3/16	4 1/8
19 3/4 20 1/2	11 3/8	9 3/8	34 1/4	33 11/16		11 13/16	11 7/16	22 1/4	13 3/8	4 11/16	5 7/16	6	6 1/4	8 7/16	4 1/4
20 1/2–21	11 7/8	9 11/16	35 5/8	34 3/4		12 3/16	11 3/4	23	14	4 13/16	5 5/8	6 3/16	6 3/8	8 11/16	4 7/16
21 –21 3/4	12 1/4	9 7/8	36 7/16	35 13/16		12 9/16	12 1/8	23 11/16	14 1/4	5 1/16	5 3/4	6 3/8	6 9/16	8 15/16	4 9/16
21 3/4–22 1/2	12 3/4	10 3/8	37 3/8	36 7/8		12 15/16	12 1/2	24 3/8	15	5 1/4	5 15/16	6 9/16	6 3/4	9 3/16	4 11/16

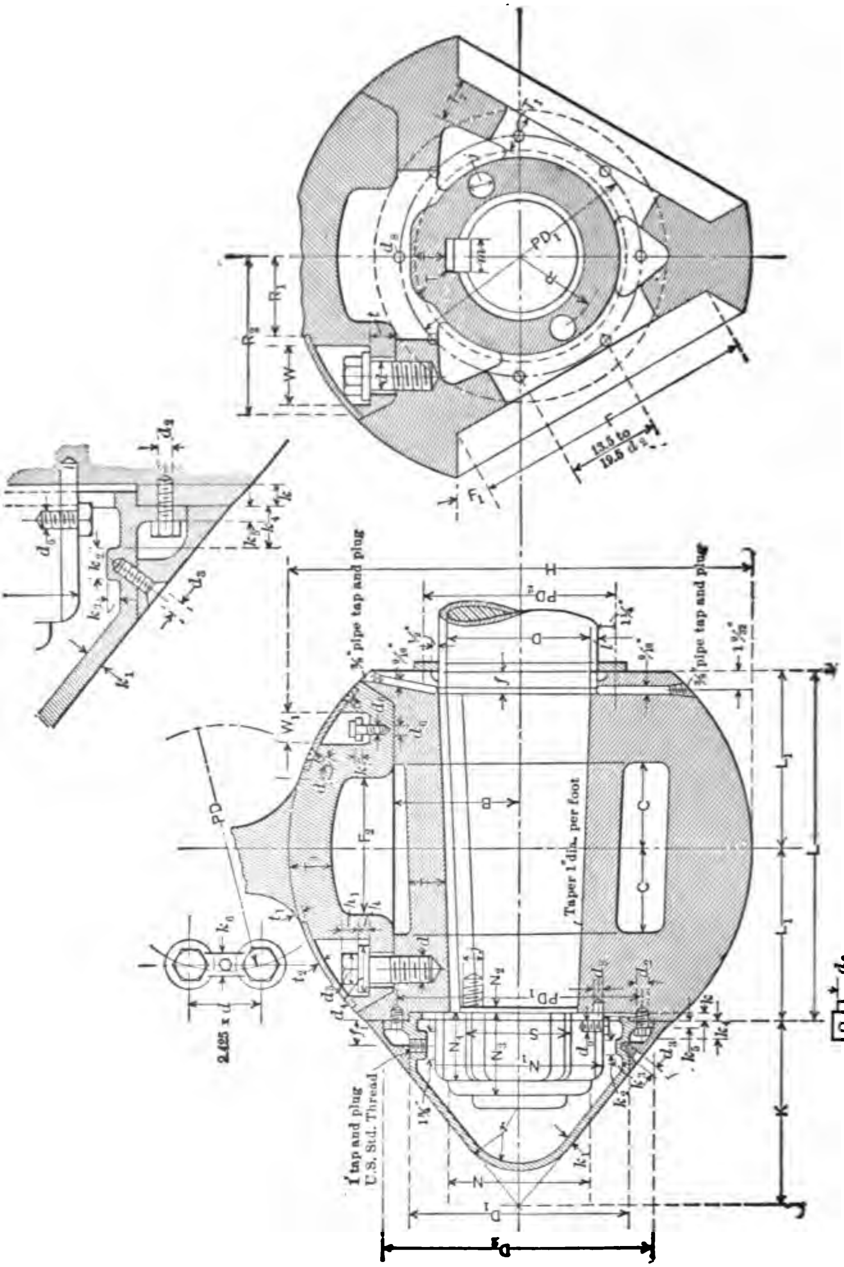

3/8" pipe tap and plug
3/8" pipe tap and plug
Taper 1" dia. per foot
13.5 to 19.5 d_2
1" tap and plug U.S. Std. Thread
2.425 x d

PLATE D

STANDARD PROPELLER HUBS—*Continued*

D	d	d_1	d_2	d_3	d_4	d_5	d_6	d_7	d_8	d_9	f	f_1	h	h_1	j	k	k_1
8 1/4	* From Bureau Design Formulas, Sec. VII, Sub. 57, sh. 5.	5/8	5/8	5/16	1.714d	1.143d	.286d	5/16	1/2	1/2	1 1/4	1 3/8	.357d	.643d	1 1/8	1/2	7/32
8 1/4–9		5/8	5/8	5/16	1.714d	1.143d	.286d	5/16	9/16	9/16	1 1/4	1 7/16	.357d	.643d	1 1/4	1/2	1/4
9–9 1/2		5/8	5/8	3/8	1.714d	1.143d	.286d	3/8	9/16	5/8	1 1/4	1 9/16	.357d	.643d	1 3/8	1/2	1/4
9 1/2–10 1/4		5/8	5/8	3/8	1.714d	1.143d	.286d	3/8	5/8	5/8	1 1/4	1 11/16	.357d	.643d	1 3/8	1/2	9/32
10 1/4–10 3/4		5/8	5/8	3/8	1.714d	1.143d	.286d	3/8	5/8	5/8	1 1/4	1 3/4	.357d	.643d	1 1/2	5/8	5/16
10 3/4–11 1/2		5/8	5/8	7/16	1.714d	1.143d	.286d	7/16	5/8	5/8	1 1/4	1 7/8	.357d	.643d	1 5/8	5/8	5/16
11 1/2–12		5/8	5/8	7/16	1.714d	1.143d	.286d	7/16	11/16	3/4	1 1/4	2	.357d	.643d	1 5/8	3/4	11/32
12–12 3/4		5/8	5/8	7/16	1.714d	1.143d	.286d	7/16	11/16	3/4	1 1/4	2 1/8	.357d	.643d	1 3/4	3/4	11/32
12 3/4–13 1/2		5/8	3/4	1/2	1.714d	1.143d	.286d	1/2	3/4	7/8	1 1/4	2 3/16	.357d	.643d	1 7/8	7/8	3/8
13 1/2–14		5/8	3/4	1/2	1.714d	1.143d	.286d	1/2	3/4	7/8	1 1/4	2 5/16	.357d	.643d	2	7/8	3/8
14–14 3/4		5/8	3/4	1/2	1.714d	1.143d	.286d	1/2	13/16	7/8	1 1/4	2 3/8	.357d	.543d	2	1	13/32
14 3/4–15 1/4		3/4	3/4	9/16	1.714d	1.143d	.286d	9/16	13/16	1	1 1/4	2 1/2	.357d	.643d	2 1/4	1	13/32
15 1/4–15 3/4		3/4	3/4	9/16	1.714d	1.143d	.286d	9/16	7/8	1	1 1/4	2 5/8	.357d	.643d	2 1/4	1 1/8	7/16
15 3/4–16 1/2		3/4	7/8	9/16	1.714d	1.143d	.286d	9/16	7/8	1	1 1/4	2 3/4	.357d	.643d	2 1/4	1 1/8	15/32
16 1/2–17 1/4		3/4	7/8	9/16	1.714d	1.143d	.286d	9/16	13/16	1	1 1/4	2 7/8	.357d	.643d	2 1/4	1 1/8	15/32
17 1/4–18		3/4	7/8	5/8	1.714d	1.143d	.286d	5/8	15/16	1	1 1/4	2 15/16	.357d	.643d	2 1/2	1 1/4	1/2
18–18 1/2		3/4	7/8	5/8	1.714d	1.143d	.286d	5/8	1	1	1 1/4	3	.357d	.643d	2 1/2	1 1/4	1/2
18 1/2–19 1/4		3/4	7/8	5/8	1.714d	1.143d	.286d	5/8	1	1 1/8	1 1/4	3 1/8	.357d	.643d	2 1/2	1 3/8	17/32
19 1/4–19 3/4		3/4	7/8	5/8	1.714d	1.143d	.286d	5/8	1 1/16	1 1/8	1 1/4	3 1/4	.357d	.643d	2 3/4	1 3/8	9/16
19 3/4–20 1/2		3/4	7/8	3/4	1.714d	1.143d	.286d	3/4	1 1/16	1 1/8	1 1/4	3 3/8	.357d	.643d	2 3/4	1 1/2	9/16
20 1/2–21		3/4	7/8	3/4	1.714d	1.143d	.286d	3/4	1 1/8	1 1/8	1 1/4	3 7/16	.357d	.643d	3	1 1/2	19/32
21–21 3/4		3/4	1	3/4	1.714d	1.143d	.286d	3/4	1 1/8	1 1/8	1 1/4	3 9/16	.357d	.643d	3	1 5/8	19/32
21 3/4–22 1/2		3/4	1	3/4	1.714d	1.143d	.286d	3/4	1 3/16	1 1/8	1 1/4	3 5/8	.357d	.643d	3	1 5/8	5/8

D	k_2	k_3	k_4	k_5	k_6	k_7	l	m	r	r_1	r_2	t	t_1	t_2
8 1/4	13/16	7/16	1 1/8	7/16	.857d	.286d	Shaft dia. D above { 7 1/2"–10" inc. l = 1/2"; 10"–12 1/2" inc. l = 5/8"; 12 1/2"–15" inc. l = 3/4" } Shaft dia. D above { 15"–17 1/2" inc. l = 7/8"; 17 1/2"–20" inc. l = 1"; 20"–22 1/2" inc. l = 1 1/8" }	From Bureau Design Formulas, Sec. II and VI, Sub. 5, sh. 45.	3 5/16	13/16	9/16	1 3/8	7/32	1/8
8 1/4–9	15/16	1/2	1 1/4	1/2	.857d	.286d			3 5/8	7/8	5/8	1 1/2	1/4	1/8
9–9 1/2	1	1/2	1 5/16	1/2	.857d	.286d			3 7/8	7/8	5/8	1 5/8	1/4	1/8
9 1/2–10 1/4	1	9/16	1 3/8	9/16	.857d	.286d			4 1/8	1	3/4	1 3/4	9/32	5/32
10 1/4–10 3/4	1 1/8	9/16	1 1/2	9/16	.857d	.286d			4 3/8	1	3/4	1 7/8	5/16	5/32
10 3/4–11 1/2	1 1/8	5/8	1 9/16	5/8	.857d	.286d			4 5/8	1	3/4	1 15/16	5/16	5/32
11 1/2–12	1 1/4	5/8	1 11/16	5/8	.857d	.286d			4 7/8	1 1/8	13/16	2 1/16	11/32	3/16
12–12 3/4	1 1/4	11/16	1 3/4	11/16	.857d	.285d			5 1/8	1 1/8	13/16	2 3/16	11/32	3/16
12 3/4–13 1/2	1 3/8	3/4	1 7/8	3/4	.857d	.286d			5 3/8	1 3/16	7/8	2 1/4	3/8	3/16
13 1/2–14	1 7/16	3/4	1 15/16	3/4	.857d	.286d			5 5/8	1 3/16	7/8	2 3/8	3/8	3/16
14–14 3/4	1 1/2	13/16	2	13/16	.857d	.286d			6	1 5/16	15/16	2 1/2	13/32	7/32
14 3/4–15 1/4	1 9/16	7/8	2 1/8	7/8	.857d	.286d			6 3/16	1 5/16	15/16	2 9/16	13/32	7/32
15 1/4–15 3/4	1 5/8	7/8	2 1/4	7/8	.857d	.286d			6 7/16	1 3/8	1	2 11/16	7/16	7/32
15 3/4–16 1/2	1 11/16	15/16	2 5/16	15/16	.857d	.286d			6 11/16	1 3/8	1	2 13/16	15/32	7/32
16 1/2–17 1/4	1 3/4	15/16	2 3/8	15/16	.857d	.286d			6 13/16	1 1/2	1 1/16	2 7/8	15/32	1/4
17 1/4–18	1 13/16	1	2 7/16	1	.857d	.286d			7 1/4	1 1/2	1 1/16	3	1/2	1/4
18–18 1/2	1 7/8	1	2 9/16	1	.857d	.286d			7 1/2	1 5/8	1 1/8	3 1/8	1/2	1/4
18 1/2–19 1/4	1 15/16	1 1/16	2 5/8	1 1/16	.857d	.286d			7 3/4	1 5/8	1 1/8	3 1/4	17/32	1/4
19 1/4–19 3/4	2	1 1/16	2 3/4	1 1/16	.857d	.286d			8	1 11/16	1 1/4	3 3/8	9/16	9/32
19 3/4–20 1/2	2	1 1/8	2 13/16	1 1/8	.857d	.286d			8 1/4	1 11/16	1 1/4	3 7/16	9/16	9/32
20 1/2–21	2 1/8	1 3/16	2 7/8	1 3/16	.857d	.286d			8 1/2	1 13/16	1 5/16	3 9/16	19/32	9/32
21–21 3/4	2 3/16	1 3/16	3	1 3/16	.857d	.286d			8 3/4	1 13/16	1 5/16	3 5/8	19/32	9/32
21 3/4–22 1/2	2 1/4	1 1/4	3 1/16	1 1/4	.857d	.286d			9	1 7/8	1 3/8	3 3/4	5/8	5/16

CHAPTER XVIII

AEROPLANE PROPELLERS. DESIGN, MATERIALS AND CONSTRUCTION

In designing aeroplane propellers, as in the design of those for the propulsion of ships, there are six variables to be taken into account. These variables are

1. The speed of flight;
2. The power required;
3. The number of revolutions of the propeller;
4. The allowable diameter of the propeller;
5. The pitch of the propeller, which is dependent on 1, 2, 3, 4;
6. The projected area ratio of the propeller.

In determining these variables, 1 is arbitrarily fixed. To obtain 2, the mean gross flying weight of the machine must be furnished. Having given this flying weight, which should include all weights on board, the machine, when flying at the designed speed, experiences a total resistance to its horizontal motion of approximately, one-sixth to one-fourth of its weight, or

$$P = \frac{\text{Gross weight}}{6}.$$

The effective horse-power, e.h.p., necessary to overcome this resistance is

$$\text{e.h.p.} = \frac{P \times v \times 5280}{60 \times 33{,}000} = \frac{P \times v}{375},$$

where v equals the speed of flight in miles per hour.

The effective horse-power thus obtained is that which is necessary for horizontal flight only, at the designed speed. For climbing purposes and for rapid turning purposes when turning from an up-wind to a down-wind direction of flight when the

wind is high, an excess of power over that necessary to deliver the effective horse-power as obtained above, must be provided. This excess power should amount to approximately 35 per cent above that required for the normal speed of flight.

Suppose the gross flying weight of a machine = 1800 lb. and the normal speed of flight be seventy miles per hour, then

$$C=\frac{1800}{6}=300,$$

$$\text{e.h.p.}=\frac{300\times 70}{375}=56.$$

The factor 6 is variable, however, and for very high speeds should be taken as low as 4.

Suppose that a propeller delivering a propulsive efficiency of 70 per cent on shaft horse-power can be fitted, then using the same notation as for hydraulic propellers with the exception of the propulsive coefficient which with aeroplane propellers equals $\frac{\text{e.h.p.}}{\text{S.H.P.}_d}$,

$\text{S.H.P.}_p=\frac{56}{.7}=80$ and the total power to be provided = $\text{S.H.P.}_{d\text{Max.}}=80\times 1.35=108.$

Number 3 of the variable elements is usually fixed by the the design of the engine and by the amount of speed reduction that is desired to be installed between engine speed of revolutions and the propeller speed.

Number 4 is limited by conditions of necessary clearances fixed by the aeroplane itself.

To obtain the necessary power, the e.h.p. being known or estimated, the proper pitch and projected area ratio, a sheet of design curves of the same general character as those used in hydraulic propeller design, has been prepared. These curves have been derived from the performances of four aeroplane propellers tested out at the United States Aviation School at Pensacola, Florida. They can not claim the same amount of accuracy, however, as can those for hydraulic propulsion as

unfortunately up to the present date there have not been developed any means of accurately measuring the power of the engine and the actual thrust of the propeller while the aeroplane is in actual flight.

In designing aeroplane propellers the designer should always be provided with a full-throttle variable brake curve of shaft horse-power and revolutions, in order to insure the ability of the engine to turn the propeller at the desired revolutions under any conditions of resistance that may occur.

Description of the Design Sheet, No. 26

On this Sheet are shown:

1. Curve of $\text{S.T.}_D = \dfrac{\text{S.H.P.} \times 33{,}000}{P \times R \times \dfrac{\pi D^2}{4} \times 144}$.
2. Curve of $\text{T.S.} = \pi D \times R$.
3. Curve of $1 - S$.
4. Curve of $\text{P.C.} = \dfrac{\text{E.H.P.}}{\text{S.H.P.}}$.
5. Curve of Log A where $A = \overline{\text{Speed}'}$.
6. Curve of $\text{S.T.}_D \div (1 - S)$.
7. Curve of Log A—0 to 110 knots.
8. Curve of Log A—110 to 180 knots.

The nomenclature is similar to that used in the computation forms for hydraulic propellers and for lack of evidence to the contrary, it is assumed that the laws governing variations of power and revolutions for hydraulic propellers apply equally as well to those operating in air.

There is a radical difference, however, between hydraulic and aeroplane propellers due to the difference in the projected area ratios which are used for the two types. While with hydraulic propellers the projected area ratios used range from about .2 to .6 and have a propulsive efficiency decreasing as the projected area ratio, tip-speed and indicated thrust increase, the aeroplane propeller has its projected area ratio between zero and .2; it is

thus located in the portion of the propeller range wherein the propulsive coefficient increases with the tip-speed, thrust and projected area ratio, referring always to the Basic Condition of Design as given by the Design Sheet. With hydraulic propellers as the projected area ratio increases the apparent slip also increases for the basic condition, and this same variation of apparent slip is seen to occur with aeroplane propellers.

With aeroplane propellers there is no such correction for slip block coefficient in the estimate of apparent slip as exists with hydraulic propellers. In other words, the air ship is treated as a phantom ship and the value of $1 \div S$ remains at a constant value for the basic condition of each value of P.A. ÷ D.A.

The Design Sheet as shown is for three-bladed propellers, the same method of correction for two- and four-bladed propellers being used as in the cases of hydraulic propellers of like number of blades.

In the application of the Design Sheet, the same method of computation may be employed as in the case of hydraulic propellers. In the example here given an alternate method is used. In the first step a constant value of P.A. ÷ D.A. and varying values of e.h.p. ÷ E.H.P. are used, while in the second step when that value of e.h.p. ÷ E.H.P. giving the nearest value to the desired revolutions with the maximum value of P.A. ÷ D.A. has been ascertained, this value of e.h.p. ÷ E.H.P. is retained constant and the problem solved for varying values of P.A. ÷ D.A.

Care must be taken that the projected area ratio does not become too large in order that the blade widths do not become excessive and deform easily under thrust. The Design Sheet carries the projected area ratio of the three-bladed propeller to .12, the table on the same sheet extending it to .17, and the limitation of these values is given in Table XVIII.

Table XVIII

Constant P.A. ÷ D.A. = .2

$P \div D$.	P.A. ÷ D.A.		
	3 Blades.	2 Blades.	4 Blades.
.0	.2	.133	.266
.1	.198	.132	.264
.2	.196	.131	.262
.3	.194	.1293	.2586
.4	.1915	.1277	.2554
.5	.189	.126	.252
.6	.186	.124	.248
.7	.1826	.1217	.2434
.8	.1793	.1195	.2390
.9	.1755	.117	.2340
1.0	.172	.1146	.2292
1.1	.1677	.1118	.2236
1.2	.1635	.109	.218
1.3	.1592	.1061	.2122
1.4	.1548	.1032	.2064
1.5	.1502	.1002	.2004
1.6	.146	.0972	.1944
1.7	.141	.0940	.1880
1.8	.1362	.0908	.1816
1.9	.1315	.0876	.1752
2.0	.127	.0846	.1692

For a 4-bladed wheel, for maximum efficiency, the total projected area ratio of the projected surface outside the .2*D* circle should never exceed .2.

Problem

Aeroplane fitted with an engine to give 125 S.H.P. at 1300 revolutions when flying at a speed of 90 miles an hour. The maximum diameter of propeller that can be carried is 8 ft. Determine pitch, projected area ratio and propulsive efficiency of a propeller to meet these conditions, the propeller to be two-bladed.

FIRST STEP

P.A.÷D.A (assumed)	.111	.111	.111	.111
$\frac{2}{3}$ P.A.÷D.A	.074	.074	.074	.074
e.h.p.÷E.H.P.	.4	.6	.8	1.0
$S.H.P._p = S.H.P._d$	125	125	125	125
Z (Sheet 21)	.4144	.231	.1009	0
$S.H.P. = S.H.P._p \times 10^Z$	325	213	158	125
P.C. for $\frac{2}{3}$ P.A.÷D.A.	.70	.70	.70	.70
E.H.P	227.5	149.1	110.6	87.5
e.h.p.	91	89.46	88.48	87.5
v	90	90	90	90
$P\times D = \frac{\pi \times S.H.P.\times 389}{S.T._D \times T.S.}$	85.26	55.88	41.45	32.79
$S.T._D$ (Sheet 26)	.1048	.1048	.1048	.1048
T.S (Sheet 26)	44450	44450	44450	44450
D	8′	8′	8′	8′
P	10.66	6.985	5.181	4.099
$V = \frac{P\times T.S.\times(1-S)}{\pi D \times 88}$	140.1	91.81	68.1	53.88
$1-S$	.654	.654	.654	.654
V in knots $= \frac{V\times 5280}{6080}$	121.7	79.73	59.14	46.79
v in knots $= \frac{90\times 5280}{6080}$	78.34	78.34	78.34	78.34
Log A_V (V in Knots). (Curve X, Sheet 21)	5.231	4.85	4.67	4.62
Log A_v (v in knots). (Curve X, Sheet 21)	4.84	4.84	4.84	4.84
$s = S\frac{S.H.P._d \times A_V}{S.H.P.\times A_v}$	.3274	.2078	1851	.2085
$R_d = \frac{v\ (\text{Miles})\times 88}{P\times(1-s)}$	1105	1431	1876	2441

Plotting these results, using Pitch for abscissas and e.h.p. and R_d as ordinates, curves are obtained (Fig. 62), from which the propeller characteristics are obtained. They are

Diameter	$=8'$;
Pitch	$=8'.25$;
P.A.÷D.A. (2Blades)	$=.074$;
S.H.P.	$=125$;
Total S.H.P.	$=125\times 1.35=168.75$;
v	$=90$ miles;

R	$=1300;$
e.h.p.	$=90;$
P.C.	$=\dfrac{\text{e.h.p.}}{125}=.72.$

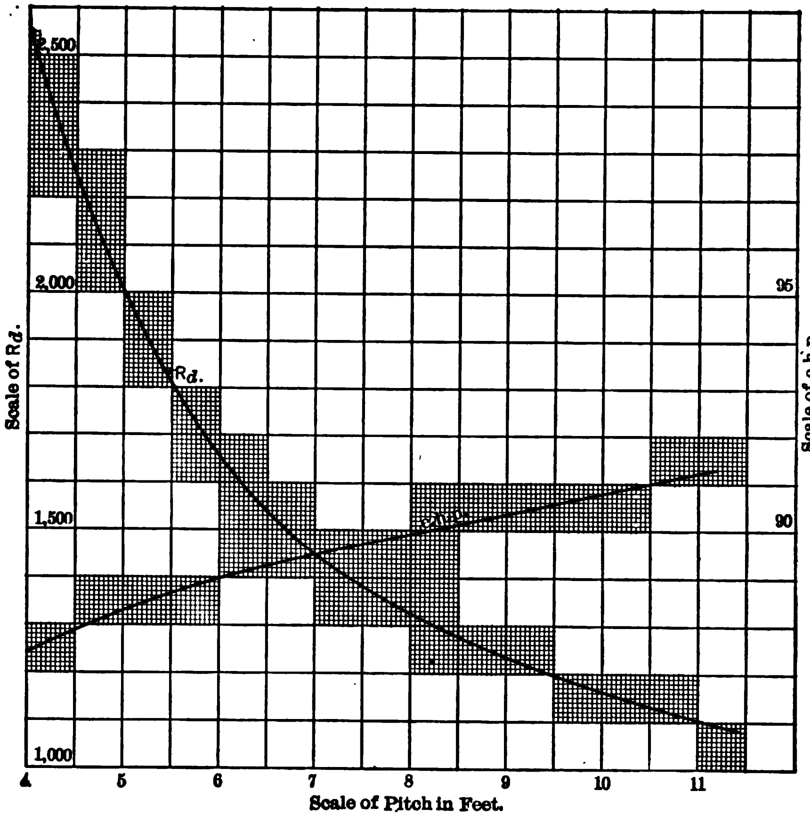


FIG. 62.

Should it be desired to increase the projected area ratio of the propeller in an attempt to obtain a higher propulsive efficiency, any load factor lower than that corresponding to the above propeller, whose load factor is somewhere between $\dfrac{\text{e.h.p.}}{\text{E.H.P.}}$ $=.5$ and $=.6$, may be taken and used as constant and the second step undertaken, as follows:

SECOND STEP

e.h.p.÷E.H.P.	.4	.4	.4	.4
P.A.÷D.A.	.111	.114	.117	.12
$\frac{2}{3}$ P.A.÷D.A.	.74	.76	.78	.8
$S.H.P._p = S.H.P._d$	125	125	125	125
Z	.4144	.4144	.4144	.4144
S.H.P.	325	325	325	325
P.C. (for $\frac{2}{3}$ P.A.÷D.A.)	.70	.702	.704	.706
E.H.P.	227.5	228.48	229.94	231.08
e.h.p.	91	91.4	91.98	92.43
D	8	8	8	8
$P\times D = \dfrac{\pi \text{S.H.P.}\times 389}{\text{S.T.}_D \times \text{T.S.}}$	85.26	79.33	73.69	68.47
P	10.66	9.916	9.211	8.558
$\text{T.S.} = 515{,}980\left(\dfrac{\text{P.A.}}{\text{D.A.}}\right)^{1.12}$	44450	45680	46992	48343
S.T._D	.1048	.1096	.1147	.12
$1-S$ for (P.A.÷D.A.)	.654	6545	.655	.6555
$V = \{P \times \text{T.S.} \times (1-S)\} \div (\pi D \times 88)$	140.1	134.05	128.19	122.62
V (knots)	121.7	116.4	111.3	104.1
v	90	90	90	90
v (knots)	78.34	78.34	78.34	78.34
A_V (knots)	5.231	5.187	5.143	5.077
A_v (knots)	4.84	4.84	4.84	4.84
s	.3274	.2958	.267	.229
R_d	1105	1134	1173	1200

The increase in P.A.÷D.A. has not been sufficient to carry the revolutions to 1300 as required, and it would be necessary to go to still wider blades of lower pitch to reach those revolutions unless it were decided to try a load factor e.h.p.÷E.H.P. between that of the first step propeller and the second step.

Should the value of P.A.÷D.A. be such as to be beyond the limits of the Design Sheet, then the various factors can be found as follows:

$$\text{S.T.}_D = 4.4\left(\frac{\text{P.A.}}{\text{D.A.}}\right)^{1.7},$$

$$\text{T.S.} \ = 519{,}580\left(\frac{\text{P.A.}}{\text{D.A.}}\right)^{1.12},$$

$1-S$ is of same value as at P.A.÷D.A. = .12.

Log A_{V} for values beyond 180 miles is found by means of the equations given for the same purpose under hydraulic propellers.

Variations in the computations produced by the change from two- to three- or four-bladed propellers.

Where three-bladed propellers are desired, and it is always preferable to use them, rather than two-bladed, on account of their smaller diameter and smoother running, the P.A.÷D.A. as taken from the Design Sheet will be those of the propellers derived, and all the data including the P.C. must be taken for those values of P.A.÷D.A

The only change in formulas which occurs is in that for $P \times D$, which becomes

$$P \times D = \frac{\pi \times \text{S.H.P.} \times 291.8}{\text{S.T.}_D \times \text{T.S.}}.$$

In the case of four-bladed propellers, this latter formula becomes

$$P \times D = \frac{\pi \times \text{S.H.P.} \times 252.41}{\text{S.T.}_D \times \text{T.S.}},$$

while all of the data with the exception of P.C. are taken for the various values of P.A.÷D.A. used, the actual projected area ratio of the propeller will be $\frac{4}{3}$ (P.A.÷D.A.) and the P.C. corresponding to this full projected area ratio must be used.

Case of Full Load and Full Diameter

In the foregoing cases, the propeller would have a large amount of reserve power and the full power of the engine could be put into it without any trouble being experienced. A propeller can be obtained directly from the design chart which will fit the full power of the engines at the revolutions and speed of flight expected and will have a very much reduced reserve capacity. In computing such a propeller, no diameter need be assumed, as the computation determines not only the pitch, projected area ratio and the propulsive efficiency but also the diameter.

Such problems are denoted as problems of "full diameter," and the method of procedure will be shown by the following computation for a full diameter three-bladed propeller:

P.A.÷D.A.	.7	.8	.9	1.0	1.1	1.2
T.S.	26500	30900	35150	39400	43800	48340
$1-S$	.834	.757	.696	.6645	.654	.6555
S.T.$_D$	.0476	.06	.0737	.088	.1032	.12
V (Estimated Speed). (Miles)	99.7	99.7	99.7	99.7	99.7	99.7
$P\times R=\dfrac{V\times 88}{1-S}$						
S.H.P. (Power of Eng.)	170	170	170	170	170	170
P.C.	.694	.706	.717	.726	.735	.744
E.H.P.	118	120	122	123	125	126
$D=\sqrt{\dfrac{291.8\times \text{S.H.P.}}{\text{S.T.}_D\times P\times R}}$	9'.953	8'.446	7'.307	6'.534	5'.986	5'.558
$P=\dfrac{P\times R\times \pi D}{\text{T.S.}}$	12'.41	9'.952	8'.233	6'.879	5'.691	5'.424
$R=\dfrac{\text{T.S.}}{\pi D}$	847.5	1165	1530	1919	2357	2468

In all of these problems, however, the work may follow the same forms as are used for hydraulic propellers, substituting S.T.$_D$ for I.T.$_D$, S.T.$_D\div(1-S)$ for I.T.$_D\div(1-S)$ and S.H.P.$_a$ for I.H.P.$_a$, and a problem worked out by this method is now given.

Shaft horse-power, revolutions and expected speed, (S.H.P.$_a$, R$_a$ and v) given, to find propeller.

Data Given

Gross load of plane = 5200 lb.
Useful load on plane = 1460 lb.
Estimated speed = 95 statute miles = 82.5 knots.
Required climb = 3900' in ten minutes.
Revolutions = R_a = 1625.
Shaft horse-power of engine = S.H.P.$_a$ = 400.
Maximum diameter of propeller that can be carried = 8' 4" = 8'.33.
Propeller to be four-bladed.

Power expended in climbing $=\frac{5200\times3900}{10\times33000}=61.46$ e.h.p.

Estimated propulsive efficiency (assumed) $=.70$.

Shaft horse-power expended in climbing $=\frac{61.46}{.70}=88$.

Shaft horse-power available for speed of advance while climbing $=400-88=312$.

Speed of advance while climbing $=v_1$.

$v_1^3 : \overline{95}^3 :: 312 : 400$.

$v_1=87.38$ statute miles.

COMPUTATION FOR PROPELLER

D (assumed)	8'.33	8'.33	8'.33	8'.0	8'.0
v (stat. miles)	95	95	95	95	95
v (knots)	82.5	82.5	82.5	82.5	82.5
e.h.p.÷E.H.P. (assumed)	.6	.7	.8	.6	.7
Z (Sheet 21)	.231	.161	.1009	.231	.161
S.H.P.$_d$	400	400	400	400	400
S.H.P. = S.H.P.$_d\times10^Z$	680.9	579.5	504.6	680.9	579.5
$v\div V$ for $\frac{\text{e.h.p.}}{\text{E.H.P.}}$ (Sheet 22)	.838	.882	.92	.838	.882
V (knots)	98.45	93.54	89.68	98.45	93.54
S.T.$_d\div(1-S)$ (Sheet 26)	.2483	.2224	.202	.2692	.2411
P.A.÷D.A. for S.T.$_d\div(1-S)$	.14375	.134	.130	.150	.1436
$\frac{4}{3}$ P.A.÷D.A.	.192	.179	.170	.200	.191
P.C. for $\frac{4}{3}\frac{\text{P.A.}}{\text{D.A.}}$ (Sheet 26)	.77	.767	.766	.77	.77
E.H.P. = S.H.P.×P.C.	524.3	444.5	386.5	524.3	446.2
e.h.p.	315	311	309	315	312
* e.h.p. (estimated necessary)	293	293	293	293	293
T.S. for $\frac{\text{P.A.}}{\text{D.A.}}$ (Sheet 26)	59180	54700	52998	62064	59110
$1-S$ for $\frac{\text{P.A.}}{\text{D.A.}}$ (Sheet 26)	.655	.655	.655	.655	.655
$P=\frac{\pi\times D\times101.33\times V}{\text{T.S.}\times(1-S)}$	6'.735	6'.923	6'.85	6'.168	6'.153
Log A_V (Sheet 21)	5.03	4.985	4.95	5.03	4.985
Log A_v (Sheet 21, Line X)	4.88	4.88	4.88	4.88	4.88
$s=S\,\frac{\text{S.H.P.}_d\times A_V}{\text{S.H.P.}\times A_v}$	.2863	.3033	.3213	.2863	.3033
$R_d=\frac{101.33\times v}{P\times(1-s)}$	1739	1733	1798	1899	1950
* e.h.p. $=\frac{95\times5280\times5200}{4.5\times60\times33000}$.					

D (assumed)	8′.0	7′.75	7′.75	7′.75
v (stat. miles)	95	95	95	95
v (knots)	82.5	82.5	82.5	82.5
e.h.p.÷E.H.P. (assumed)	.8	.6	.7	.8
Z (Sheet 21)	.1009	.231	.161	.1009
$S.H.P._d$	400	400	400	400
$S.H.P.=S.H.P._d \times 10^Z$	504.6	680.9	579.5	504.6
$v \div V$ for $\frac{\text{e.h.p.}}{\text{E.H.P.}}$ (Sheet 22)	.92	.838	.882	.92
V (knots)	89.68	98.45	93.54	89.68
$S.T._d \div (1-S)$ (Sheet 26)	.219	.2868	.257	.2334
P.A.÷D.A. for $S.T._d \div (1-S)$	.134	.155	.145	.140
$\frac{4}{3}$ P.A.÷D.A.	.179	.207	.193	.187
P.C. for $\frac{4}{3}\frac{\text{P.A.}}{\text{D.A.}}$ (Sheet 26)	.767	.77	.77	.768
E.H.P.=S.H.P.×P.C.	387.1	524.3	444.5	387.6
e.h.p.	310	315	312	310
* e.h.p. (estimated necessary)	293	293	293	293
T.S. for $\frac{\text{P.A.}}{\text{D.A.}}$ (Sheet 26)	54700	64390	59756	57454
$1-S$ for $\frac{\text{P.A.}}{\text{D.A.}}$ (Sheet 26)	.655	.655	.655	.655
$P=\frac{\pi \times D \times 101.33 \times V}{T.S.\times(1-S)}$	6′.374	5′.759	5′.883	5′.879
Log A_V (Sheet 21)	4.95	5.03	4.985	4.95
Log A_v (Sheet 21, line X	4.88	4.88	4.88	4.88
$s=S\frac{S.H.P._d \times A_V}{S.H.P.\times A_v}$	.3213	.2863	.3033	.3213
$R_d=\frac{101.33 \times v}{P \times (1-s)}$	1933	2034	2040	2091
* e.h.p. $=\frac{95 \times 5280 \times 5209}{4.5 \times 60 \times 33000}$.				

All revolutions obtained are too high. To reduce them hold the value of e.t.÷ E.T. corresponding to any chosen condition of $\frac{\text{e.h.p.}}{\text{E.H.P.}}$ and $\frac{v}{V}$, in this case .55 and 838 as the propeller obtained for those conditions and with a diameter of 8′ 4″, had low revolutions combined with maximum efficiency, constant and reduce the values of $\frac{\text{e.h.p.}}{\text{E.H.P.}}$ and $\frac{v}{V}$ by coming down along the line of this $\frac{\text{e.t.}}{\text{E.T.}}$ on Sheet 22. as follows:

e.t.÷E.T.	.715	.715	.715
e.h.p.÷E.H.P.	.55	.5	.47
S.H.P.$_d$	400	400	400
Z	.27	.3135	.342
S.H.P.	780	823.2	879.2
P.A.÷D.A. (as before)	.144	.144	.144
$\frac{4}{3}$ P.A.÷D.A.	.192	.192	.192
P.C. for $\frac{4}{3}\frac{\text{P.A.}}{\text{D.A.}}$	.77	.77	.77
E.H.P	600.6	634	676.9
e.h.p.	315.4	317	318.2
$v \div V$ for $\frac{\text{e.t.}}{\text{E.T.}}$ and $\frac{\text{e.h.p.}}{\text{E.H.P.}}$	.768	.698	.655
v (knots)	82.5	82.5	82.5
V	108.7	118.2	126
T.S. for $\frac{\text{P.A.}}{\text{D.A.}}$ (Sheet 26)	59180	59180	59180
$1-S$ for $\frac{\text{P.A.}}{\text{D.A.}}$ (Sheet 26)	.655	.655	.655
$P=\frac{\pi \times D \times 101.33 \times V}{\text{T.S.} \times (1-S)}$	7'.434	8'.086	8'.617
Log A_V	5.12	5.215	5.28
Log A_v	4.88	4.88	4.88
$s=S\frac{\text{S.H.P.}_d \times A_V}{\text{S.H.P.} \times A_v}$	.322	.3625	.3943
$R_d=\frac{101.33 \times v}{P \times (1-s)}$	1659	1622	1602

Take that propeller promising 1622 revolutions as the one to be used. Its characteristics are as follows:

Blades	4
Diameter	8′ 4″
Pitch	8′ 1″

Total projected area ratio outside of .2 radius of propeller .192.

Standard $\frac{\text{P.A}}{\text{D.A.}}$ form	.144
S.H.P	400
R	1625
v	95 statute miles.

Construction of Aeroplane Propellers

Material furnished by Mr. Spencer Heath, American Propeller Co.

The aeronautical screw propeller or air screw, strange as it may seem, was in point of theory and conception, at least, the forerunner of the hydraulic screw.

Like nearly all things aeronautic, in its own day this invention was scorned and neglected, particularly by the man of science, and afterwards virtually forgotten, so much so that when the same device many years later was employed for marine propulsion it was hailed and received as wholly new.

In their basic principle all screws are the same for whatever purpose used. They differ only in their proportions, form and material. These differences are due to the different nature of materials through which the screws act. A machine screw acting through a previously prepared nut is a special case of a member progressing along an inclined plane, the movement of the member being along a helical curve instead of a straight line. A wood screw and a corkscrew, however, make their own " nut " through the cork or wood in which they act. These screws, in common with the machine screw, have an axial advance in one revolution substantially equal to the distance between turns of the " thread," known as the pitch of the screw. The air and hydraulic screw propellers are like the cork and wood screw in that they form their own " nut " through the air or water, but they differ markedly in that the fluid medium through which they pass has no great stability and yields to the screw in such manner that commonly it does not advance its full pitch in one revolution. This yielding of fluid is entirely analogous to the yielding of the water to the bending of an oar or paddle. The resistance with which the fluid yields in the sternward direction creates a fulcrum for the oar and measures the propulsive impulse of oar or screw. The magnitude of this impulse depends upon the density of the fluid medium and the sternward velocity with which it is caused to move.

The differences between air and hydraulic screw propellers may be said to be the reflex of the differences between air and

water. Air is thinner, lighter, larger, swifter than water. Air propellers are longer and thinner of blade, lighter in weight of material, larger in proportion to duty and swifter in velocity of rotation and of flight.

Almost all air propellers are made of wood. In general appearance, except for width of blade, they resemble hydraulic propellers. The number of blades may be two, three or four. The widest part of the blade is usually at about six-tenths to seven-tenths of its radius. The maximum width of the developed arc, measured for zero pitch and on the circular arc, averages about one-twelfth the diameter of the screw. The thickness of blade near the hub is very great but diminishes rapidly to about half-blade length and then gradually to the end. The side of the blade facing rearwardly (the driving face), is of true and constant pitch and may be slightly concave in its wider and mid-radius portions and flat or slightly convex at its narrowed end and near the hub. The forward facing side of the blade (the back of the blade), is convex in all parts and the greatest thickness of any section is about one-third the distance from the entering to the trailing edge. Nearly all air propellers have the general characteristics already mentioned. In blade outline, however, there is wide diversity. Some designers prefer to approximate a slender ellipse; others prefer the slender ellipse with squared ends on the blades; others approximate a semi-ellipse, the axis of the ellipse proceeding radially from the axis of rotation and forming the trailing edge of the blade. Among these various forms no special preference is known. Some of them are laid out with a view to placing the centers of gravity of all the blade sections in one radial line; some with the aim of having the center of pressure on each blade section lie in the same straight line.

As in hydraulic propellers, it is desirable to adhere to a standard form of blade if a rapid solution of the propeller problem is to be obtained.

In all the above forms of blade a common property obtains: The deflection of the blade under load is accompanied by more or less increase of angle in its most effective parts, thus aug-

menting the pitch. This gives the blade a sort of unstable pitch which may introduce heavy strains and resistances to the turning of the propeller at the moment when highest turning speed is required. In order that the pitch may remain unaffected by bending or deflection of the blades under load or increase of load it is necessary to dispose the centers of pressure of the sections farthest from the hub on a line curving somewhat rearwardly in the direction of the trailing edge. Variable pitch propellers are designed by so far extending the rearward curvature of the blades that the application of working blade pressure will institute a torsional action on the entire blade causing its pitch to increase or diminish in response to variation in pressure. In this torsion design for variable pitch the portion of wood employed in the curved trailing edge of the blade is steamed and bent so that the grain of wood parallels the curved edge of blade. In this process a slight compression is given to the fibres which greatly increases the endurance of the thin edge portions of the blade.

Nearly every kind of wood has been used in propellers. Walnut and mahogany have long been favorite in Europe. The experience of one noted builder rules out all wood that was not quarter sawed and points to American quartered white oak as the surpassing material from every standpoint, the particulars of which need not be detailed here.

It is almost needless to say that the wood for air propellers should be selected and treated with utmost care. The boards are sawn to 1 in., rough dressed to $\frac{7}{8}$ and finish dressed to $\frac{13}{16}$ or $\frac{3}{4}$-inch thickness.

The propellers are built up by five or ten laminations according to size. The laminations are laid out on the boards and sawed to outline, care being taken to avoid all defects in wood and to have the grain and density of wood as nearly similar as may be at opposite ends of the same piece. In the better and preferred practice, however, the laminations for each single blade are laid out separately and carefully weighed, matched and balanced against each other. They are then selected in pairs (or in trios for three-bladed propellers) and their hub

ends securely glued together in highly efficient joints of very large glue-contact area. Only by this method is it possible to make the blades of the same propeller uniform in respect to weight, grain texture and yielding of the wood under stress.

When the separate laminations have been prepared and surfaced to required thickness they are slightly roughened by tooth planing, warmed over steam coils and assembled together with the best of hide stock glue and firmly clamped. The entire gluing process is carried on in a room kept at 100° F. After eighteen to twenty-four hours the clamps are removed and center hole in hub bored roughly to size. The propeller is now hung for about ten days to dry. It is then put through a machine which at one operation faces both sides of the hub and bores out the center hole to finish size. After being faced and bored the propeller is "outlined" in a machine that profiles the hub and edges of the blades all to exact size and shape by means of a rotary cutter following a form which has the precise outline of the blades.

From the outlining machine the propeller progresses to one of the duplicators. In this machine the work is clamped in definite relation to a rigid fixed form having the same shape as the blades. On the carriage of the machine there is a roller which traverses the surface of the form and guides a high speed cutter in a manner to remove nearly all surplus wood from the rough propeller. The carriage is self feeding and self reversing and the bed and other parts of the machine, including the form, are made duplex in order to secure continuous operation of the cutter, the work being removed and renewed at each end of the machine in turn while the carriage is operating uninterruptedly at the other end.

After the duplicating process the propeller again dries for a few days after which it is carefully surfaced and balanced by hand and then forwarded to the sanding machine. After sanding there is careful inspection before proceeding further, and careful examination of balance, pitch and tracking of blades, hub dimensions, etc. The inspection itself is an elaborate process requiring special appliances, etc., of various kinds.

From inspection the propeller passes to the finishing department. Here it is first treated with silex filler, then with primer, and lastly with various coats of high test waterproof spar varnish. This is the usual finish. For certain United States Army work five applications of hot linseed oil and a final rubbing with prepared wax are required. During the entire varnishing process the propeller is kept carefully balanced on a steel mandrel resting on sensitive parallel ways. Without this very few propellers could pass final inspection.

On final inspection the utmost attention is given to every detail. Balance must be absolutely perfect in all positions; the blades must track, that is must follow each other in the same path, within .03 in.; the pitch of the blades checked at three points must not vary from the standard by more than 2 per cent nor from each other more than 1 per cent. These limits are only to allow for possible changes in the wood during the finishing process after the first inspection.

A few words should be said as to the number of blades: For training work and all ordinary work, provided a sufficient diameter can be swung, two blades are usually preferred. For expert flying and for high-powered machines in which there is a restricted diameter of propeller in proportion to power applied, three and four blades are required. As to the relative merits of three and four blades there are no conclusive data. It is known, however, that in numerous instances the three-bladed screw, even though having less diameter, shows marked superiority over the two-bladed *in every particular*. The three-bladed propeller is also noted for its peculiar jointing and fitting of the ends of the laminations together where they form the hub. This hub is trebly laminated over its entire area with the material so disposed as to direction of grain, etc., that it makes without doubt the strongest hub that can be built in any propeller regardless of the number of blades. In repeated cases of wreck and accident all the blades of these propellers have been wholly demolished leaving the hubs always intact.

After final inspection the propellers are usually packed in standardized pine or white cypress boxes with screwed-on covers

and heavy battens and iron-bound ends. A center bolt clamps the propeller between battens in the top and bottom of the box and felt-lined pillow blocks formed to the shape of the screw secure it firmly in place.

CHAPTER XIX

CONTENTS OF ATLAS

In the Atlas accompanying this text will be found:

1. Barnaby Chart of Propeller Efficiencies. Sheet 16.
2. Chart for correction of Block Coefficients. Sheet 17.
3. Chart for estimation of Appendage Resistance. Sheet 18.
4. Chart for thrust deduction. Sheet 19.
5. Chart of Design, maximum thrust, Basic Conditions. Sheet 20.
6. Chart of Design, Estimate of Revolutions and of Z for power. Sheet 21.
7. Chart of Load Limitation. Sheet 22B.
8. Chart of Thrusts. Sheet 22.
9. Chart of values I.T.$_D \div (1-S)$. Sheet 23.
10. Chart of values of (P.A.÷D.A.)×E.T.$_p$. Sheet 24.
11. Chart of Standard Forms of Projected Area Ratios, curves showing relation between projected—and helicoidal-area ratios, and table of multipliers to use in laying down standard forms. Sheet 25.
12. Table of Hull and Propeller Characteristics for a large number of vessels, giving the nominal block coefficients, the standard block coefficients as corrected by line X and the coefficient of immersed amidship section, and the slip-block coefficient as corrected for location of propeller. Sheets 12, 13, 14, 15.

These tables also include the performance of the different propellers, including indicated thrusts per square inch of disc and per square inch of projected area, the indicated thrust being taken as equal to the shaft thrust÷.92, where shaft horse-power was originally given. They also include the propulsive coefficient on the bare hull, the appendage multiplier as obtained from

Sheet 18, and the resultant propulsive coefficient on the hull with all appendages.

13. A large number of cuts showing one-half of the projected areas of many of the blades studied and whose performance is given in the foregoing tables; also the blade sections tangent to the hubs. In order to compare these forms more readily, all propellers have been reduced to a common diameter, the sections being reduced in the same scale, and all the projected areas are arranged symmetrically around the center-line of the blade normal to the axis of the hub. Sheets 1 to 11.

Upon these projected areas are shown, in dotted lines, the standard form of blade projection, Sheet 25, having approximately the same area.

A comparison of these forms will show that the majority of the most successful propellers have projected area forms approximating very closely to the proposed standard forms of projection, while of those not having the standard form, the most successful are slightly broader at the tips of the blades. Those which are narrower at the tips than the standard show higher tip-speeds and higher slips than the charts will give.

Sheet 26. Design Sheet for Aeroplane Propellers.

Sheet 27. Blade Form Sheet for Aeroplane Propellers.

Sheets 28, 29, 30, 31, 32. Examples in geometry and draughting of the hydraulic propeller.

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